THE AMERICAN SOCIETY OF
MECHANICAL ENGINEERS

TRANSACTIONS

VOLUME 30

DETROIT MEETING
NEW YORK MEETING
1908

NEW YORK
PUBLISHED BY THE SOCIETY
29 West 39th Street
1909

CORRECTION IN TRANSACTIONS, VOL. 29

Ball Bearings, by Henry Hess, Transactions, Vol. 29, p. 447, caption of Fig. 14, Relation of Compression and Load for the Three Tests, Item 2 should read :

"2 balls ⅝ in. diameter and flat disc. Compression according to Hertz $\frac{\delta}{2} = 0.0000805 \sqrt[3]{\frac{2}{P}}$"

CORRECTION IN TRANSACTIONS, VOL. 26

The Forcing Capacity of Fire-tube Boilers by F. W. Dean, Transactions, Vol. 26, p. 93, in table, Par. 6, the last six items refer to fire-tube boilers instead of water-tube boilers.

Prof Wm C Hoad
6-25-44

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THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
1908

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PAST PRESIDENTS

Thurston, R. H.	1880–1882	Died Oct. 25, 1903
Leavitt, E. D.	1883	Cambridge, Mass.
Sweet, John E.	1884	Syracuse, N. Y.
Holloway, J. F.	1885	Died Sept. 1, 1896
Sellers, Coleman	1886	Died Dec. 28, 1907
Babcock, George H.	1887	Died Dec. 16, 1893
See, Horace	1888	New York, N. Y.
Towne, Henry R.	1889	New York, N. Y.
Smith, Oberlin	1890	Bridgeton, N. J.
Hunt, Robert W.	1891	Chicago, Ill.
Loring, Charles H.	1892	Died Feb. 5, 1907
Coxe, Eckley B.	1892–1894	Died May 13, 1895
Davis, E. F. C.	1894	Died Aug. 6, 1895
Billings, Charles E.	1895	Hartford, Conn.
Fritz, John	1896	Bethlehem, Pa.
Warner, Worcester R.	1897	Cleveland, O.
Hunt, Charles Wallace	1898	New York, N. Y.
Melville, George W.	1899	Philadelphia, Pa.
Morgan, Charles H.	1900	Worcester, Mass.
Wellman, S. T.	1901	Cleveland, O.
Reynolds, Edwin	1902	Milwaukee, Wis.

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1908

Dodge, James M.	1903	Philadelphia, Pa.
Swasey, Ambrose	1904	Cleveland, O.
Freeman, John R.	1905	Providence, R. I.
Taylor, Fred. W.	1906	Philadelphia, Pa.
Hutton, F. R.	1907	New York, N. Y.

[According to the Constitution, Article C 27, the five Past Presidents who last held the office shall be members of the Council, with all the rights, privileges and duties of the other members of the Council.]

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1908

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Note.—Numbers in parentheses indicate length of term in years that the member has yet to serve.

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1908

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D. S. Jacobus
A. P. Trautwein
G. T. Voorhees
Philip De C. Ball
E. F. Miller

On Society History

John E. Sweet
H. H. Suplee
Charles Wallace Hunt

On Affiliated Societies

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On Government Advisory Board on Fuels and Structural Materials

P. W. Gates
W. F. M. Goss
Geo. H. Barrus

Note.—Numbers in parentheses indicate length of term in years that the member has yet to serve.

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1908

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Chas. E. Sargent
F. P. Thorp
E. P. Coleman
C. L. Straub
W. T. Magruder
Arthur L. Rice

SUMMARY OF MEMBERSHIP

December 31, 1908

UNITED STATES

	Membership		Membership
Alabama	10	Montana	10
Alaska	2	Nebraska	6
Arizona	2	Nevada	3
Arkansas	2	New Hampshire	11
California	64	New Jersey	233
Colorado	28	New Mexico	2
Connecticut	142	New York	923
Delaware	16	North Carolina	13
District of Columbia	29	North Dakota	1
Florida	1	Ohio	243
Georgia	15	Oklahoma	3
Hawaii	3	Oregon	9
Illinois	221	Pennsylvania	420
Indiana	65	Porto Rico	3
Iowa	9	Rhode Island	69
Kansas	11	South Carolina	2
Kentucky	8	Tennessee	12
Louisiana	27	Texas	10
Maine	13	Utah	5
Maryland	30	Vermont	8
Massachusetts	301	Virginia	26
Michigan	86	Washington	10
Minnesota	15	West Virginia	9
Missouri	53	Wisconsin	76

Total membership in the United States	3260
Address unknown	1

FOREIGN COUNTRIES

	Membership		Membership
Africa	14	Holland	1
Australia	9	India	1
Austria	2	Italy	1
Belgium	6	Japan	9
Canada	43	Mexico	10
Central America	3	Norway	1
China	3	Russia	3
Cuba	4	San Domingo	1
England	42	Scotland	3
Finland	1	South America	11
France	9	Sweden	5
Germany	9	Switzerland	3

Total foreign membership	194

SUMMARY OF MEMBERSHIP

Geographical

December 31, 1908

Foreign membership	194
Membership in United States	3260
Address unknown	1
Total	3455

By Grades

December 31, 1908

Honorary members	15
Members	2322
Associates	357
Juniors	761
Total membership	3455

ATTENDANCE AT MEETINGS 1908

The following figures show the attendance at the several meetings of the Society during 1908

January	14	New York			196
February	11	" "			223
March	10	" "			274
April	14	" "			372
May	12	" "			159
June	23–26	Detroit (Semi-annual)	Members	272	
			Guests	445	717
October	13	New York			178
November	10	" "			234
December	1–4	" " (Annual)	Members	738	
			Guests	310	1048

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DETROIT, NEW YORK AND MONTHLY MEETINGS, 1908

MINARD LAFEVER HOLMAN

President of The American Society of Mechanical Engineers, 1908

Minard Lafever Holman, the President of the Society during 1908, was born June 15, 1852, in Mexico, Maine. He is descended from a New England family, his paternal great grandfather being Daniel Holman of Maine, one of the "Minute Men" of the Revolution. In 1859, Mr. Holman's parents removed to St. Louis, where his father, John Henry Holman, enlisted in the Civil War. After the close of the War, he was appointed Military Governor of Eastern North Carolina, and commissioned Brevet General.

Mr. Holman's early education was gained in St. Louis, Missouri, and Chelsea, Massachusetts, and he was graduated from Washington University in 1874 with the degree of C.E., and in 1905 the honorary degree of Master of Arts was conferred upon him by his Alma Mater.

His first engagement after graduation was in the supervising architect's office of the United States Treasury at Washington, D. C., where he remained two years, after which he spent some time in Tennessee, and upon returning to St. Louis entered the engineering offices of Messrs. Fladd & Smith. In 1877, when the senior member of the firm was elected President of the Board of Public Improvements of St. Louis, Mr. Holman entered the Water Department of the city as a draftsman, advancing until he became principal assistant in the department ten years later. He then resigned to accept the appointment of Chief Engineer of the Missouri Street Railway Company, when it was preparing to change its system from horse to cable traction. He had just entered this work when the Honorable David R. Francis, at that time Mayor of St. Louis, tendered him the appointment as Water Commissioner, in charge of the entire water system of the city. Mr. Holman immediately took up the work of extending and modernizing the system, the city expending for this purpose about $10 000 000.

During the twelve years that he held this office, he caused to be installed a new low-service pumping plant and six large settling basins, and extended an intake to the main channel of the Mississippi

River. He installed new pumping stations, equipped with up-to-date machinery, which saved the city 80 per cent of fuel. He introduced the system of paying a bonus to builders for high efficiencies in pumping engines, which has been followed in the department since and has resulted in the making of several famous high duty records. It was under Mr. Holman's administration as commissioner that a division of the water distribution was made, so that a part of the city was supplied from one point and the balance from another. He also inaugurated the experiments in water sedimentation and purification by the aid of chemicals. In 1897 he visited Europe for the purpose of making a study of foreign filtration systems.

Mr. Holman served on the Board of Public Improvements of St. Louis, which has charge of the public engineering and structural work of the city, and as Chairman of the Committee on City Lighting had charge of the undergrounding of the electric wires, and the preparation of the specifications and letting of contracts for the lighting of streets, alleys and public buildings.

During his public service, Mr. Holman always stood firm against political control of important positions, holding that efficiency, and not political activity, should determine the applicant's fitness for appointment and retention. As a result, his department reached a high state of efficiency.

Upon the expiration of his third term as Water Commissioner, Mr. Holman decided to devote himself to private engineering practice and opened an office in St. Louis. A year later he accepted the appointment of General Superintendent of the Missouri Edison Electric Co., which operated the largest lighting and power plants in the city. He continued this work until the sale of all the local plants to the North American Syndicate in 1904, when he formed a partnership with Mr. John A. Laird for private practice in hydraulic and mechanical engineering.

Mr. Holman's success in his administration in the St. Louis Water Department established him as an authority in this field, and he has since been called into consultation regarding problems in many other cities, among which are Kansas City, Cincinnati, Omaha and Denver. He served on the Board of Appraisers of Denver, which determined the value of the water system of that city and fixed schedules of water rates. He has also been called into consultation from Canada, Mexico and Germany.

Mr. Holman is a member and Past-president of the Engineers Club of St. Louis; an honorary member of the American Water Works

Association; a member of the American Society of Civil Engineers since 1884, and Vice-president in 1905–1906; a member of the Loyal Legion by inheritance from his father; and a member of several Masonic fraternities. He has served two terms as Vice-president of this Society, 1894–1896 and 1903–1905, and was chairman of the local entertainment committee at the St. Louis Convention of the Society in 1896.

Mr. Holman is a member and president of the Third Congregational Church of St. Louis.

CONFERRING HONORARY MEMBERSHIP ON MR. CARNEGIE

The inaugural banquet of the Engineers' Club was made the occasion for conferring Honorary Membership in the Society upon Andrew Carnegie. The banquet was held at the new club house, No. 32 West 40th Street, Monday evening, Dec. 9, 1907.

Andrew Carnegie, Mem. Am. Soc. M. E., Samuel L. Clemens and Thomas A. Edison, Mem. Am. Soc. M. E., were the guests of honor. T. Commerford Martin, President of the Club, was toast master. In his initial address, he paid tribute to the originality of Mr. Carnegie's thought in the simultaneous recognition of the equal importance of the social with the technical side of engineering, as exemplified in his gift of a club house for engineers and a building for the engineering societies.

Mr. Martin offered a toast, "Not to the greatest iron master of the age; not to the creator of a thousand and one libraries; not to the liberal endower of research and education; not to the rewarder of heroic self-sacrifice,—but to our benefactor, our intimate and our associate, our comrade and our friend,—Andrew Carnegie."

Mr. Carnegie responded in an address in which he spoke of the value of recognizing the personal equation. He said in part:

"All knowledge is contributive to every part of other knowledge. In whatever department of engineering you are, you all contribute to each other. It is the greatest benefit to come in contact with men in different departments of your business, and I look upon this club as one of the most potent forces for producing splendid results that has been projected."

He spoke of the possibilities of the club for making permanent friendships, and closed with the words from Hamlet:

> "I think myself in nothing else so happy,
> As in a soul remembering my dear friends,'

Mr. Martin then introduced John Fritz, Hon. Mem. and Past President, Am. Soc. M. E., paying tribute to "the love and sweetness, the purity and charity" of his character.

Mr. Fritz made the presentation speech of Honorary Membership in this Society to Mr. Carnegie, and gave a few reminiscences of the early days of engineering, touching upon the difficulties which brought about a closer association of prominent engineers, resulting in the formation of the engineering societies, and afterward in the closer association of the engineering societies themselves in the magnificent home provided for them by Mr. Carnegie. He said in part, "the engineer is the most important person upon the world's stage today, whether the drama be peace or war. Mr. Carnegie, in erecting these magnificent buildings and dedicating them to this use and purpose, has recognized the work and worth of the engineer, and placed him in a position most justly his. The kindly feelings existing between the engineering societies promise well for the hope that through the influence of the engineers, nations may be drawn into closer bonds, and by our united efforts, all flags entwined in an everlasting emblem of peace. May this be the glory of the engineer, and the crowning glory of the Twentieth Century."

Mr. Martin then received from the President and the Secretary of The American Society of Mechanical Engineers the certificate of honorary membership, and presented it to Mr. Carnegie. The certificate was signed by Thomas A. Edison, Alex. C. Humphreys, E. R. Archer, John A. Brashear, Charles H. Morgan, John Fritz, W. R. Warner, John E. Sweet, Thos. Fitch Rowland, S. T. Wellman, Charles Wallace Hunt, and Charles H. Haswell.

Mr. Carnegie, in accepting the certificate, generously acknowledged his indebtedness to the engineer, and expressed heartfelt appreciation of the honor conferred upon him.

Mr. Martin then introduced Samuel L. Clemens (Mark Twain), who gave a characteristically humorous speech on simplified spelling,—placing the blame for "this new disaster upon the human race" upon Mr. Carnegie. He said that Mr. Carnegie had attacked orthography at the wrong end; having attacked the symptoms and not the disease, the real disease being in the alphabet. John Foord spoke of the Scot in America, and said of them that they had performed a useful function in the association of the practical with the ideal. "It will take time to show how great are the ideals of Mr. Carnegie. His idea of beneficence is not to blunt the strength of the people who are the beneficiaries, but to nerve them to new effort."

Mr. William H. Fletcher, Chairman of the Building Committee of the Club, reported as follows: Cost of the property \$265 000. Cost of the house \$545 597.80, of which \$450 000 was received from Mr.

Carnegie. Cost of furnishing the Club $57 285.31. Making the total cost of the building and land $867 883.11.

The total indebtedness of the Club: mortage to the Bowery Savings Bank of $110 000, and bonds sold to members $200 000, making a total of $310 000, leaving an equity of $557 883.11.

Mr. Martin, in a closing speech, acknowledged the valuable gifts to the club from prominent manufacturing concerns, for its power house equipment.

MONTHLY MEETINGS

HELD IN NEW YORK
JANUARY 14, FEBRUARY 11, MARCH 10, APRIL 14,
MAY 12, 1908

REGULAR MONTHLY MEETINGS

THE JANUARY MEETING

The January meeting was held in the Engineering Societies' Building on Tuesday evening, January 14, at 8:15 o'clock, John W. Lieb, Jr., Vice-president, presiding.

"Car Lighting" was the subject of a paper presented by R. M. Dixon, Mem. Am. Soc. M. E. It was discussed by George R. Henderson, H. K. Brooks, Lamar Lyndon, B. P. Flory, R. E. Bruckner, W. E. Ver Planck, W. D. Young and George L. Fowler.

THE FEBRUARY MEETING

Members and others interested in the formation of a Gas Power Section assembled in the rooms of the Society in the Engineering Societies building at 7:30 p.m. on February 11.

The meeting was called to order, and Dr. C. E. Lucke was chosen Chairman and H. H. Suplee, Secretary.

The Secretary of the meeting then read the rules for the formation of Professional Sections prepared by a special committee of the Society consisting of F. R. Hutton, R. H. Fernald, A. C. Humphreys, H. H. Suplee and F. W. Taylor. These rules having been previously approved by the Council, the prospective members of the section also approved them and proceeded to organize under them.

In substance they stated that a section must be officered by members of the Society and that all ordinary expenses will be borne by the Society. Extraordinary activities may be carried on as a section may elect, but at its own expense.

The intent of the Society is to give opportunity for any of its members to specialize without being obliged to form a new body. It further provides that persons not members of the Society may enroll as affiliates of a section. Thus the Society places at the disposal of sections or affiliated bodies the benefit of the organization of the Society.

An executive committee of five was chosen as follows: the Chair appointed Jesse M. Smith and Albert A. Cary, a committee to

prepare nominations; R. H. Fernald, G. I. Rockwood, F. H. Stillman, F. R. Low and H. H. Suplee were nominated, and no further nominations being offered, a vote was taken and they were unanimously elected.

The name of Dr. Chas. E. Lucke was then presented for Chairman of the Section, and he was unanimously elected to serve one year from December 6, 1907.

The meeting then adjourned to reassemble in the main auditorium where the following professional papers were presented and discussed, Prof. C. E. Lucke presiding.

"A Continuous Gas Calorimeter" (No. 1187 in this volume), by Dr. Chas. E. Lucke. Mem. Am. Soc. M. E.

This included a description of the Junker calorimeter, and of several modifications which had been made to render such devices continuous in their readings, followed by a description of the improved calorimeter invented by the author of the paper, in which the ratio of the flow of gas and water was maintained constant, enabling the calorific value of the gas to be determined at any moment by the simple inspection of temperatures. The addition of thermo-electric recording apparatus to the calorimeter enables it to furnish continuous records. The paper was illustrated by lantern slides.

"Recent French Experimental Gas Turbines" by H. H. Suplee, Mem. Am. Soc. M. E.

This was an exhibition of lantern slides of the 300 h.p. gas turbine constructed by the *Société des Turbomoteurs* of St. Denis, from the designs of Messrs. Armengaud and Temale, as well as of the high-pressure multiple-turbine air compressor designed by Rateau for use in connection with gas turbine experiments.

'Gas Engine and Producer Guarantees" by Dr. Chas. E. Lucke, Mem. Am. Soc. M. E.

After considerable discussion, in which the importance of greater definiteness in guarantees was generally admitted and urged, it was voted that the Chair be directed to appoint a committee, of which Dr. Lucke should be a member, to examine the subject and report to the Section.

"A Gas Electric Car" by H. G. Chatain, Mem. Am. Soc. M. E.

Mr. Chatain described the car built from his designs by the General Electric Company at Schenectady, and now undergoing tests. This includes an eight-cylinder gasolene engine, direct-connected to a dynamo, furnishing current for electric motors on the trucks. The engine is of 125 horse power, and the car weighs about 31 tons, the

engine having driven the car at a speed of more than 50 miles an hour.

THE MARCH MEETING

The monthly meeting for March, held in the Engineering Societies' Building, on March 10 at 8:15 o'clock, was well attended, Prof. L. P. Breckenridge, Vice-president of the Society presiding.

A paper, "The Steam Path of the Turbine," was presented by Dr. Charles P. Steinmetz, Mem. Am. Soc. M. E.

It was discussed by Prof. C. H. Peabody, S. L. Kneass, Prof. S. A. Reeve and H. E. Longwell.

THE APRIL MEETING

The Conservation of our Natural Resources

The April meeting of the Society, held in the Engineering Societies building on April 14, at 8.15 o'clock, was devoted to the Conservation of our Natural Resources, in response to the invitation by the President of the United States to coöperate for conserving the natural resources of our country.

The Society extended a general invitation to members of the entire engineering profession to attend this meeting and prominent representatives of the four national engineering societies were seated on the platform with the speakers of the evening.

John W. Lieb, Jr., Vice-President of the Society, was chairman of the meeting and in his opening remarks extended a welcome to the distinguished speakers, to the officers and representatives of sister engineering societies and to the guests.

Four addresses were given as follows:

"The Conservation of the Waters and Woods," by Dr. W J McGee, Secretary of the Inland Waterways Commission, representing the United States.

"The Conservation of the Nation's Fuel Supply," by Dr. W. F. M. Goss, Dean of the College of Engineering, University of Illinois.

"The Conservation of Stream Flow, Water Power, and Navigation," by Dr. George F. Swain, Director of the Department of Civil Engineering, Massachusetts Institute of Technology.

"The Relation of the Engineer to the Body Politic," by Dr. Henry S. Pritchett, president of the Carnegie Foundation for the Advancement of Teaching..

At the close of the addresses a number of lantern slides were shown by Dr. McGee.

Several letters were read by the Secretary, among which was the following from the President of the United States:

THE WHITE HOUSE, WASHINGTON

April 13, 1908.

MR. CALVIN W. RICE, *Secretary, The American Society of Mechanical Engineers, 29 West 39th Street, New York.*

My dear Mr. Rice:

I regret that time did not permit me when you were here to tell you how gratified I am with the zeal and interest which the members of the engineering profession are taking in the movement to conserve our natural resources.

It is the duty of all our citizens to exercise foresight in these matters, but it is peculiarly the duty of the engineer, who is in fact a trustee of the forces of nature. Above all men the engineer is responsible for seeing that our natural resources are not wasted and that their development is for the common good and for the continuance of the welfare of our citizens.

I was particularly pleased with the resolutions passed by your Council and indeed with the coöperation of all engineering bodies in bringing this matter forcibly to the attention of all the people. I am sorry that I can not be with you on Tuesday evening to hear the addresses, but I extend to all the engineers my heartiest good wishes.

Sincerely yours,

[Signed] THEODORE ROOSEVELT

The following telegram was received from President Schurman of Cornell University, recently appointed by the Governor of New York as one of the delegates to accompany him to the Washington conference:

ITHACA, N. Y., April 14, 1908.

SECRETARY CALVIN W. RICE *29 West 39th Street, New York.*

Regret I cannot attend engineers meeting today. I attach greatest importance to your deliberations on conservation of the country's natural resources.

J. G. SCHURMAN.

Mr. Lieb then announced that the purpose of the meeting was to discuss the nature and extent of our resources; the demands which have been made upon them in the past, and which may be expected in the future; whether our use of them has been economical or wasteful; and suggestions in the direction of future public policies which may lead to a more efficient utilization and a wiser conservation of our natural resources. He reminded the audience that our first President, the immortal Washington, himself an engineer of ability, was at the time of his election president of the Delaware and Chesapeake Canal

Company, and now the President of the United States following in the footsteps of his predecessor has created a commission on inland waterways whose duty it is to formulate a broad policy for the protection and control of our water supply.

Then followed the first address, by Dr. W J McGee, Secretary of the Inland Waterways Commission, of which the following is an abstract:

THE CONSERVATION OF THE WATERS AND WOODS, BY DR. W J MCGEE

Dr. McGee pronounced the subject of conservation, touching, as it does, the conditions of the perpetuity of our Nation, the most important we have ever had to consider. He said what the country specially needs today is a mental and moral revolution and awakening by the individual to a sense of obligation for the preservation of the gifts of nature, not only for the present generation, but for all the generations to come. He spoke especially of the water as one of the important resources to be conserved. The average rainfall over the mainland of the United States is about thirty inches, or about 200 000-000 000 000 cu. ft. per year, which would make about ten lower Mississippi. Just as soon as our population and industries have increased to such an extent as to consume this annual rainfall, we shall have reached the limit of our development, unless by that time human ingenuity has devised some way, recognized by only a few of our scientists at the present time, of producing water in addition to that we receive from the heavens.

Dr. McGee said that about three-fifths of the rainfall is evaporated and serves to temper the air, producing dews, fogs, and regulating the temperature; about one-fifth passes deeply into the earth, or is consumed in various chemical combinations on the earth's surface; the other fifth flows down to the sea. It is for the conservation of this one-fifth that the people of the United States should be most concerned.

There is no diminution in the rainfall or the precipitation on the surface, but too much water flows away on the surface, producing torrents in the streams and the chain of evils following.

Our injudicious treatment of water, our failure to appreciate its importance, is greatly influencing navigation. The Ohio river is an example. At Cincinnati it has a range from low water to flood water of more than 50 ft., so that it is extremely difficult to maintain terminals in such a manner as to render the Ohio an effectively navi-

gable stream. These floods are constantly increasing because of the deforestation at the headwaters of mountains and foothills in which the waters first gather, so that the soil is no longer a great sponge, drinking in the water and giving it back slowly, but sheds it with a rapidity which gathers it in torrents, and the torrents in turn take millions of tons of soil on the way and deposit them in the channels of the stream, an obstruction to navigation. The navigability of our streams on the whole today, is much less than it was at the time of the first settlement of each district.

Dr. McGee further said: The soil is one of the assets of the earth which is only lately recognized by this nation. It is the original vegetable humus, the accumulation of centuries, even of milleniums. The Mississippi valley alone is losing every year more than 500 000-000 tons of richest soil matter through soil erosion, and the entire United States is losing somewhere between 1 000 000 000 and 2 000-000 000 tons. Considering the lowest price of this soil as a fertilizer, which cannot be less than $1 a ton, the farmers of the United States, through soil erosion alone, are paying an annual tax of between $1 000 000 000 and $2 000 000 000, which is absolutely without return.

The loss of our soil means the loss of our forests; the exhaustion of our iron ores; the exhaustion of our coals. To consider one relation between coal, iron and water, where rivers are navigable, it requires only about one-tenth as much iron to carry a given cargo of freight by water as to carry the same cargo by rail, so that improvement in our návigation will diminish the tax on our iron ores. It requires to carry each ton of freight only about one-eighth (depending of course on the speed movement) as much coal to carry the freight on water, as is required to carry the same freight by rail, so that improvement of navigation will cut down the rate of consumption of coal, at least relatively. These are among the relations that might be multiplied indefinitely.

THE CONSERVATION OF THE NATION'S FUEL SUPPLY, BY DR. W. F. M. GOSS

Abstract

This subject was presented by the speaker under four headings: The Value of Fuels, The Production of Fuels, Lack of Economy in the Use of Fuel, and How Economy of Fuel is to be Secured.

Under the first heading striking illustrations were given of the

magnitude of the mining and allied industries. In 1850 there were mined in the United States 6 000 000 tons of coal, and since this date the annual production has more than doubled every ten years, until in 1906 it reached the enormous amount of 414 000 000 tons.

In addition to these 414 000 000 tons of anthracite and bituminous coal mined, there were drawn to the surface nearly 400 000 000 000 cu. ft. of natural gas weighing over 9 000 000 tons and 126 000 000 barrels of oil weighing approximately 17 000 000 tons. Summarizing these gives the enormous total of 440 000 000 tons, an amount so great as to challenge the imagination.

Fancy it all, for example, to have the form of marketable size bituminous coal and to be dumped into a windrow of triangular cross section, piled to a height of 32 ft., with a width of base of 46 ft. Such a pile would be as high as an ordinary two-story pitched-roof house. It would contain nearly 30 tons per foot run, while the length of the windrow would be sufficient to reach from New York to San Francisco. This measures the annual rate of production at the present day only; what will be the rate 10, 50, or 100 years hence?

Under the second heading Dr. Goss recounted some of the wasteful methods of mining. The tendency in many districts is to accomplish with powder what ought to be done with pick or machine. Excessive charges are common, which increases the percentage of fine coal, in many mines never hoisted. Heavy explosions not infrequently bring down the roof of the mine over large areas of coal which under present day conditions cannot be successfully opened up again. When a good layer of coal is overlaid by one or more thinner layers which cannot be worked with the same facility as the layer below, it is common practice to take out the coal below and to permit the roof above to cave in, destroying the continuity of the thinner layers above, making subsequent mining operations impossible.

Or the coal above may be heavy and of satisfactory quality, but seamed with thin layers of shale or slate, so that the expense of separating it from its impurities makes it unattractive and it is consequently not taken out. Again, the presence of abnormal amounts of sulphur in portions of a layer of coal otherwise satisfactory may result in the partial removal of a single heavy layer, the undesirable portions being left as abandoned property. It is estimated by one high in authority,[1] whose experience has been chiefly in the Appalachian field, that approximately 50 per cent of the coal in the mines never gets to the surface.

[1] I. C. White, State Geologist of West Virginia.

In the production of oil actual waste has probably not been great, although the supply has been heavily drawn upon. The waste of natural gas, however, has been very great. In 1891 it was estimated by Mr. Gorby, State Geologist of Indiana, that the amount of gas daily consumed at the wells and actually wasted was in excess of 100 000 000 cu. ft. Since that time the wastes have been allowed to continue. Twelve years later a pipe line delivering 1 000 000 cu. ft. a day was found to be leaking six times the volume it delivered. As the supply of gas diminished, oil appeared and old gas wells were uncapped for the purpose of blowing down the pressure to promote the flow of oil. Under the influence of heavy legitimate consumption and of these wastes, the Indiana field is now, seventeen years after Mr. Gorby made the statement above referred to, practically dry. Fuel sufficient in quantity to have served generations of people was squandered in a period of two decades.

Under the third topic the speaker called attention to the fact that in every movement of fuel, from the point of production to the stack of the furnace in which it is used, loss occurs which, by the exercise of attention, may be reduced or entirely eliminated.

The greater part of the fuel supply is consumed in industrial processes, a comparatively small quantity being required to keep the nation warm. Nearly one-fourth of the total output, or about 100 000 000 tons, goes to the railroads and is for the most part consumed in locomotive fireboxes; that is, 51 000 locomotives are today burning coal at a rate which is in excess of the rate of total production 25 years ago. Approximately 20 per cent of the coal supplied locomotives is used in starting fires, or in keeping the machine hot while standing on side tracks, or is left in the firebox at the end of the run. Sixteen million tons of the annual consumption are thus accounted for. From 8 to 10 per cent of the remainder is discharged as unconsumed fuel from the stack during the operation of the locomotive and the remainder is required for the generation of steam.

If American locomotives were of a more highly developed type such as are employed for foreign service, if they were designed with compound cylinders or with superheaters, the amount of fuel required would be less and the annual coal consumption would be reduced 6 000 000 to 10 000 000 tons.

The use of coal in the manufacturing industries of the country is on the whole probably not attended by a higher degree of efficiency than that which prevails on railways. Everywhere there are evidences of bad or imperfect practices.

In pointing out how economy of coal is to be secured, Dr. Goss said the necessary steps are:

a Scientific research for the establishment of facts;

b Practical development of the facts thus developed on a scale which will convince men that there is profit, direct or indirect, in a better practice;

c Restrictive legislation which will protect the public from the competition of unscrupulous men;

d Effective inspection which will secure enforcement of laws.

The process must begin with education—not with coercion.

The speaker then showed how improvement could be secured at the mines through a better understanding of the use of explosives and regulation of their use, referring to the improved practice in England in this regard. Processes must be developed for working coal above ground whereby inferior grades may be better adapted to use. Fine coal must be washed and if this is not sufficient to give them a market they must be coked or briquetted. Slaty and sulphurous coals must be crushed and washed and processes now undiscovered must be called into service to make it profitable to hoist from the mine every ounce of available fuel. Wherever it can be shown that such processes are practicable, it is but reasonable to compel their use.

In the matter of the economical use of fuel we still have much to learn, a statement emphasized by the fact that the problem relates largely to the use of bituminous fuels, of all fuels the most difficult to handle. The task of learning how to burn efficiently and without smoke the good and bad soft coals of our country is a task assigned to the present generation, and The American Society of Mechanical Engineers constitutes the strongest single influence in this work.

It has not yet been shown as a commercial proposition how small, soft coal fires under boilers can be maintained without smoke. No laws have been deduced to define the desirable length of flameway for coals of different compositions. The possibility of a much more general application of the gas producer, both as an individual piece of apparatus and as a detail in furnace construction, are yet before us. These and many similar matters are fundamental in the development of a more economical practice.

The conservation of our fuel resources is an engineering problem and it will yield to treatment in proportion to the engineering skill which is concentrated upon it. The matter is of national importance and should receive national attention.

THE CONSERVATION OF STREAM FLOW, WATER POWER AND NAVIGATION, BY PROF. GEORGE F. SWAIN

Abstract

In opening his address Professor Swain drew a parallel between the individual and the nation, and said it is always a delicate task to induce either an individual or a people to live prudently and husband its resources. This country, like almost all others, has erred both in unduly drawing upon its natural resources beyond the power of reproduction to support the consumption, when such power exists; and also by waste and extravagance, unaccompanied by any corresponding good.

Turning to the subject of water, he said that when the country had a small population pure water could be had in unlimited quantity. But with the recklessness characteristic of the human race we allowed our streams to become polluted, and the public health endangered. Our streams should not only be preserved in volume, but in purity. Much depends, also, upon the preservation of the regularity of stream flow, and streams may be preserved and regulated in two ways:

a By the preservation of forests;

b By constructing reservoirs.

The preservation of forests appeals to us both as a source of supply of important material and as a means of affecting stream flow. Waste and fire have so reduced our originally magnificent forests that now we are told there is but 20 years' supply left.

It is improbable that forests increase the total rainfall or run-off. Indeed there are reasons for thinking that they sometimes decrease the total run-off. Trees intercept about one-fourth the total rainfall by their leaves and branches, not allowing it to reach the ground; they also absorb considerable moisture by their roots, to be evaporated through their leaves; but these losses are more than compensated for by the reduction of evaporation of water from the ground or from water surfaces, which is only about one-fourth to one-third as great in forests as in cleared ground. The forests act as a shield against the wind, cool the air and increase the relative humidity. Some writers claim that forests evaporate immense quantities of water, but as Ebermayer found the total amount of moisture in the air to be almost the same in forests as in cleared ground, we may dismiss this contention; and we furthermore have the conclusion of Risler that forests evaporate less than grass land.

Facts were also cited to prove that forests are regulators of flow and that whether they increase or decrease the total run-off is unimportant; for the value of a stream depends not upon its total flow, but upon the regularity of that flow. The humus of a forest is a great sponge which absorbs the water and gradually gives it out.

Examples were quoted of the effect of removing forests. In France a small spring flowed from a cleared area, a forest of firs was allowed to grow up, the spring broke forth in much larger volume, and for 40 or 50 years was considered the best in the neighborhood. Finally the woods were cut off, the spring dwindled, and conditions became as they were 90 years before. Becquerel, Marsh, and other writers give many examples showing that the cutting down of the forests has been followed by the drying up of springs, the lowering of lakes, and an increasing irregularity in the flow of rivers.

Hough, in his Elements of Forestry, says:

> The Khanate of Bucharia presents a striking example of the consequences brought upon a country by clearings. Within a period of 30 years this was one of the most fertile regions of central Asia, a country which, when well wooded and watered, was a terrestrial paradise. But within the last 25 years a mania for clearing has seized upon its inhabitants and all the great forests have been cut away, and the little that remained was ravaged by fire during a civil war. The consequence was not long in following and has transformed the country into a kind of arid desert. The water courses are dried up and the irrigating canals empty. The moving sands of the desert, being no longer restrained by barriers of forests, are every day gaining upon the land, and will finish by transforming it into a desert as desolate as the solitudes that separate it from Khiva.

Forests diminish the violence of floods. A wooded surface can never pour forth such deluges as flow from cleared land, or especially from unwooded mountain slopes, and the forests on these should be protected, especially as such land is not suited to agriculture.

> In the French Department of Lozere, which was among those most severely injured by the inundations of 1866, it was everywhere remarked that the ground covered with wood sustained no damage, even on the steepest slopes, while in cleared and cultivated fields the very soil was washed away and the rocks laid bare by the pouring rain.

If forests are cut down and not replaced, we must therefore expect an increasing irregularity of flow of streams. We must not, however, claim too much, or expect that by allowing forests to grow freshets can be entirely done away with. If fire follows lumbering, as is often the case, the soil is damaged and the trees cannot grow upon it for a long period.

It is not necessary to emphasize the importance of water power to an audience of engineers. Many of our industries came into existence because of it and the cities of Lawrence, Lowell and Manchester would very likely never have been founded in their present locations but for the existence there of great waterfalls.

In the five streams with their tributaries which drain the White Mountain region, it is estimated that 350 000 h.p. is utilized, this being about 20 per cent of the total utilized water power of the country, and nearly 3 per cent of all the utilized power of the country. In the Southern States the development has been also great, especially in recent years, but there are still greater possibilities. The United States Geological Survey estimates that there is a minimum of 2 800-000 h.p. generated on streams draining the Southern Appalachians, of which one-half or 1 400 000 h.p., can be economically developed. At $20 per horse power per annum this represents $28 000 000 per annum. This estimate is given simply to indicate that even relatively small injuries to water powers may result in a great injury in the aggregate.

The second means of controlling stream flow is by storage reservoirs. This method has not yet been undertaken to any large extent, though every mill pond is a storage basin. This method, however, will naturally be used more and more and the reservoirs will be located near the headwaters of the streams where the floods have their origin. Here the effect of the destruction of forests is most injurious, for from the steep unwooded slopes the rain and melting snows will carry off great quantities of earth and deposit it in the rivers and reservoirs; and the best part of the earth, fertile top soil, will be carried off in this way, not only filling up the streams, but making the soil incapable for a long time of supporting a forest growth. This silting up is especially to be feared in the Southern Appalachians, where the soil is deep to the mountain tops, but it is also of importance in the White Mountains. We spend great sums for dredging the lower reaches of our rivers, whereas the preservation of our forests would render unnecessary some of this work at least.

What should be our position as engineers on matters of this kind? Should we not exert ourselves to do our full share both in the development and conservation of our natural resources? Engineering is a noble profession Engineers as a rule are honest, intelligent, capable men. They have been trained to cope with the forces of nature and to direct them for the use and convenience of man. They have been taught to seek the truth and not to make the worse seem the

better reason. This is the age of the engineer, but the engineer has not yet come to his own. He is too often looked upon as a servant or employee, a tool for others to use, not a leader. This is perhaps to some extent the fault of the engineers themselves. Too frequently they are so much absorbed in details of their profession that they do not grasp, or seek to grasp, the broader problem of which their work is a part.

In the development and conservation of the nation's resources is it not to our own profession that the nation should look for the men, who by training and knowledge are best fitted to show the way? An individual responsibility rests upon us to take an active, interested, enthusiastic part, directing public opinion instead of being subservient to it.

THE RELATION OF THE ENGINEER TO THE BODY POLITIC, BY DR. HENRY S. PRITCHETT

Abstract

A profession as distinguished from a business implies a vocation in which not only is expert service applied for the benefit of him who uses it, but also in the interest of the state and the public. We are prone to consider the essential difference between the work of the engineer of today and the engineer of the past to lie in the greater achievement which the engineer of the present day may compass through the largeness of his enterprises and the rapidity with which great projects are carried out. This way of thinking is, I think, essentially wrong. I doubt whether the engineer of today deserves any great honor, as compared with the engineer of a thousand years ago, from the mere consideration of the size of his operations. An engineer is a man who applies the science of his time to the construction of works which have to do with the needs of his day. Measured by this standard, the engineer of today can boast very little over the engineers who raised the pyramids, who built the temples of Karnak or the hanging gardens of Babylon, in the mere matter of size.

The great difference between the accomplishments of the engineer of today and of the engineer who built the pyramids lies not in the difference of magnitude, but in the difference of purpose in which their respective tasks were undertaken. The engineer of a thousand years ago wrought as the servant of a king, did the things that the king commanded, and these things in many cases were associated

with enterprises of little moment for civilization or for improvement.

The engineer of today works also in employment, not always of a king, but of an employer or a company; but his work differs from that of the engineers of a thousand years ago in the fact that he works always in the service of mankind. The building of the pyramids was a splendid engineering feat, which always excites our admiration, but it had small significance in promoting the comfort, happiness or civilization of the people for whom they were built. There was celebrated in this city last week an event much less picturesque: the opening of the four tunnels which connect the city of New York with Long Island, and yet this latter work of the engineer has a far larger public interest and is of far higher public service.

Engineering has grown from a vocation to a profession. It has entered into the company of those great callings whose members are recognized as not only the servants of those who employ them, but as the guardians, also, of the public interest and the public honor. There can be no question that professions in our social order rise to great power only by reason by the strict sense of honor of their members and again lose relatively by a lack of observance of that sense. There is no truer word than that of Bacon, "I hold every man a debtor to his profession."

It is this standing of engineering as a profession, rather than a business, a sense of honor founded on the highest professional ideals, rather than in the exigencies of business life, which I wish to urge upon the engineers of our day.

And what is the practical significance of such a distinction in this matter which we are considering? Is the engineer to decline to carry out the wishes of his employer? Is he to set himself up as the judge of that which is fitting and right and profitable, for those who pay his salary?

The answer to these questions lies in the use of fair judgment, a sound sense of justice, and a quick appreciation of the larger public causes. The engineer is in a unique position to exercise by his advice, his suggestions, and his consciousness of the public interest, a great influence in the encouragement of justice and of wisdom. It will be a part of the honor due to his profession to look always at the larger rather than at the smaller view of development; to undertake the consideration of great enterprises rather from the standpoint of the great and unlimited future than from the standpoint of the small and limited present. In a word he will, if he be a true member of a profession, while serving loyally his employer, keep ever before the

eye not only of himself, but of those whom he serves, the honor of his own profession, the debt which he owes to it, and the service of the larger interests of humanity which these considerations require.

THE MAY MEETING

The May meeting was held in the Engineering Societies' Building on the evening of May 12, Fred J. Miller, Vice-President, presiding. The paper of the evening, upon "Clutches" (No. 1188 in this volume), was given by Henry Souther, of Hartford, Conn., Mem. Am. Soc. M. E., and a member of the Society of Automobile Engineers and of the Association of Licensed Automobile Manufacturers. There was a representative audience and the discussion was correspondingly varied and complete. Mr. Souther dealt largely with clutches as used in automobiles, and much of the discussion which followed supplemented what he had to say by descriptions of clutches developed in other fields of work. Charles R. Gabriel of New York showed a number of lantern slides of clutches used in power press work, and Frank Mossberg of Attleboro, Mass., touched upon the same class of work. E. H. Neff, New York, and E. J. McClellan, New York, took up clutches in machine tool work. Others who participated were Fred Miller, Buffalo, N. Y., J. J. Bellman, New York, H. F. J. Porter, New York, and B. D. Gray, of the American Locomotive Co., Providence, R. I.

This paper was brought up again for discussion at the spring meeting at Detroit.

No. 1187

A SIMPLE CONTINUOUS GAS CALORIMETER

By Prof. C. E. Lucke, New York
Member of the Society

All heat engines in practical operation, when supplied with commercial fuels, may have their performance expressed in pounds of coal, gallons of oil, cubic feet of gas or other fuel units per horse power hour, but as the thermal value of the different specimens of any one fuel is not constant, such a mode of defining performance fails to give the thermal efficiency of the engine or plant, and in addition, and what is more important commercially, it is impossible to judge which is the cheaper fuel.

2 The high class steam plants and large consumers have come to the point of purchasing coal in heat units instead of tons, and their specifications contain a fixed price for one quality of fuel, based on its calorific power as found by systematic and regular calorimeter tests, and carry with this clause a bonus for exceeding a fixed calorific power and a penalty for failing to reach it. The coal in any one shipment will run fairly constant in calorific value, so it is only necessary in such tests that the calorific value of a fixed number of samples be taken to evaluate the lot, in order to reduce the purchasing of coal to the basis of the purchasing of heat.

3 The coal calorimeter thus has been brought into use as a commercial measuring apparatus, as valuable in the daily work of the plant as the coal scales, and its sphere of usefulness has in consequence been extended beyond the more occasional efficiency tests of the coal fired plant.

4 In a similar manner the calorific value of a fuel gas fixes the output of the gas producer, the input energy of the gas engine, and the usefulness of public service and other gas supplies for such purposes as furnace heating, cooking and the production of light by all mantle methods. The gas calorimeter has found a useful place in tests of gas for such purposes, but as the calorific power of a gas may vary

Presented before the Gas Power Section at the Monthly Meeting (February 11, 1908), of The American Society of Mechanical Engineers.

widely and quite suddenly, such tests become of value only when made continuously, which is not possible with the ordinary intermittent instrument. And further, to be of commercial value in the purchase of heat in the form of gas instead of buying cubic feet of gas, it is necessary that the gas calorimeter be made automatic as well as continuous, or at least that it be continuous, and operative by common unskilled labor. In the everyday operation of a gas power plant, involving the gas engine and producer, the daily performance of one cannot be separated from the other without such an instrument, nor can the engine and producer operator do more than guess at the proper adjustment and handling of their apparatus.

5 The general need of such an instrument, that would be as useful to the gas maker and gas user as the steam gage is to the fireman and steam engineer, has been keenly felt, and serious effort has been made to supply the need; but as the work of different investigators has never been published it is necessary to resort to the records of the patent office to show the nature of this effort, which has not as yet resulted in the placing of such instruments on sale.

6 Prof. Hugo Junkers of the *Royal Polytechnicum* at Aix-la-Chapelle, Germany, patented in this country March 10, 1896, the gas calorimeter shown in Fig. 1, which up to the present has been the standard instrument for the determination of the heating power of gases. The principle underlying the instrument is extremely simple, and as most of the members are familiar with it, it will not be described in detail here.

7 A gas burner of the Bunsen type liberates the heat of combustion of the gas. The hot products of combustion from this burner, together with any excess air that may have been admitted to the chamber containing it, pass vertically upward through a straight flue, under the ordinary draft influence of the heated column. They then pass downward through a number of small tubes surrounded by water which, in cooling them below the atmosphere, assists in the descent of the gases.

8 Adjustments are provided in the instrument, consisting of a water-controlled valve to regulate the quantity continuously flowing through, and different gas orifices for the burner, to accommodate different gases which may vary in calorific value from less than 100 to more than 1000 B.t.u. per cubic foot, to the end that the final temperature of the issuing products of combustion may be reduced to that of the atmosphere, it being assumed that the gas is supplied at the same temperature as the atmospheric air. With corrections for

radiation, this mode of procedure results in giving to the water flowing through the instrument all the heat of combustion of the gases and none of the heat of the atmosphere which supports the combustion.

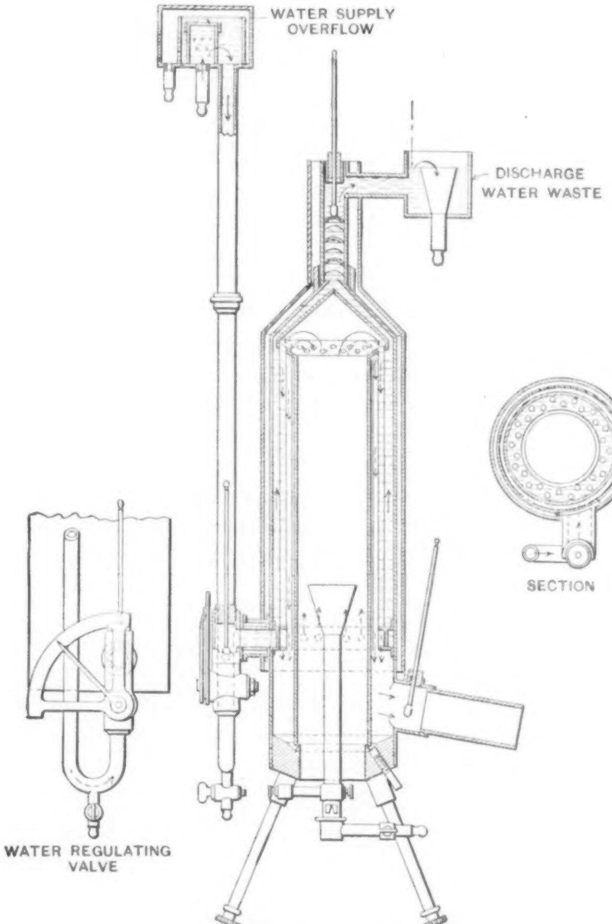


FIG. 1 JUNKERS GAS CALORIMETER

9 An accurate wet gas meter is used to measure the amount of gas burned, and a graduated liter jar, into which the discharging water can be directed at will, serves to measure the quantity of water simul-

taneously with the gas. A small gas holder is also provided to remove from the meter, as far as possible, the pressure fluctuation effects of the pipe line, permitting the gas to be measured at a constant pressure.

10 Wide fluctuations of gas pressure exist in ordinary working, especially in blast furnace plants, often varying from ten inches water pressure above to several inches below atmospheric pressure, without warning and in short periods of time. In such cases the gas supply is to be provided for by separate means, as must also be done when a suction producer, which is always under a vacuum, is under observation. The best method I have found of eliminating this variable pressure trouble has been an ordinary suction tee, discharging into a reservoir chamber with a water sealed overflow.

11 Another defect of the ordinary Junkers instrument that becomes somewhat serious with weak gas is the tendency for the gas to go out at the burner, due to the lifting action of the draft on the flame. This I have succeeded in overcoming in practically every case by placing a flat disc of copper about one-eighth inch above the burner outlet, and extending beyond the burner about one-half inch all around. The flame will then burn under the disc with no tendency to lift and is supplied with the proper amount of air on the under side.

12 A third defect, and probably the most important commercially, is that after the observations have been made of quantity of gas, quantity of water and temperature rise of the water, which require a reasonably skillful man and considerable time, the calorific power is found only by calculation as follows:

$$\text{B.t.u. per cu. ft. of gas} = \frac{\text{lb. of water}}{\text{cu. ft. of gas}} \times \text{temp. rise of water}$$

13 The instrument therefore, though found extremely valuable in test work, fails under the conditions of operation previously mentioned; is not continuous, and therefore is not as valuable commercially as it might be. It cannot be handled by a common fireman in charge of a producer, and furthermore is quite expensive.

14 It was pointed out by Junkers in 1903, and later by several others independently, that if the ratio of the water supply to the cubic feet of gas burned in the same time could be kept constant, then the calorific power would be directly proportional to the temperature rise of the water. A patent was issued to Junkers in England in 1904 and on October 19, 1906, in the United States, ten years after his original patent, describing an instrument to maintain this proportionality and to record the differences in temperature, thus making

the instrument continuous and automatic. Fig. 2 shows what was proposed by Junkers to accomplish this result. To the gas meter there is now added a water meter, and the two meters are geared together.

15 I doubt whether any water meter and gas meter can be made that will accomplish the desired result when one drives the other, or when both are driven from an outside source, until new principles in meter operation are discovered; and the additional meter adds to the expense. The rise in temperature of the water is found directly

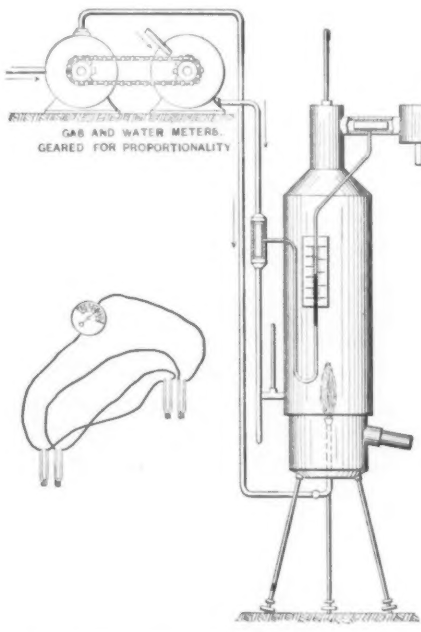


Fig. 2 Junkers Calorimeter, with Water Meter, which is Geared to Gas Meter

by a sliding scale extending over the stems of the two thermometers and graduated the same as these thermometers; the zero to be applied to the low temperature and the rise to be read off directly opposite the high temperature. By introducing on this sliding scale the constant of proportionality fixed by the meters, a second scale may be added, giving directly the British thermal units per cubic foot of gas.

16 To make the instrument recording, or readable from a distance, Junkers proposed a thermo-couple and a milli-voltmeter,

which may be either of indicating or of recording type, and will give directly the temperature difference between the hot and coid junctions.

17 A second means of attaining proportionality was proposed by Schutte and Koerting and a patent covering it was granted them on January 2, 1906. This involves the principle of displacing gas by the water to be used, both directly and indirectly, two forms being shown. In the direct form, Fig. 3, a pair of small tanks is arranged above the gas heater or calorimeter and provided with numerous valves and ports to control the flow of water and gas. Water admitted to one of these chambers displaces gas from it, the gas having previously been drawn in while the water was running out to the instrument. The various valves are operated by the rise and fall of these two chambers, due to the weight of water they contain, with the net result that the water and the gas are taken alternately from the right and left hand chambers, the gas being forced to the burner by the filling of one chamber with water, and the water being supplied to the instrument by the emptying of the other chamber simultaneously, drawing in a gas supply.

18 This principle of displacement produces intermittent action and an element of uncertainty in momentary ratios of water to gas, because the rate of gas supply depends on the rate with which a tank fills with water from a constant head supply, while the rate of water supply depends on the rate with which a similar tank discharges this water under a variable head. In order that the momentary rate of filling and emptying two similar but separate tanks may be the same, there is involved a very complicated problem in practical hydraulics which is not here solved.

19 The second form of apparatus for the attainment of proportionality proposed by these men, shown in Fig. 4, includes a pair of piston pumps and is free from the objection noted above, but is also very complicated and costly. The principle involved in both of them however—the attainment of proportionality by displacing the gas with the water—is is extremely interesting, although in practical application it is somewhat difficult to reduce to simplicity and positiveness.

20 Another interesting new point in this case is the means proposed for recording the temperature differences after the proportionality is established. Instead of using the thermo-couple proposed by Junkers, a means is shown depending upon the expansion of a liquid which is really a differential liquid thermometer. Into the water

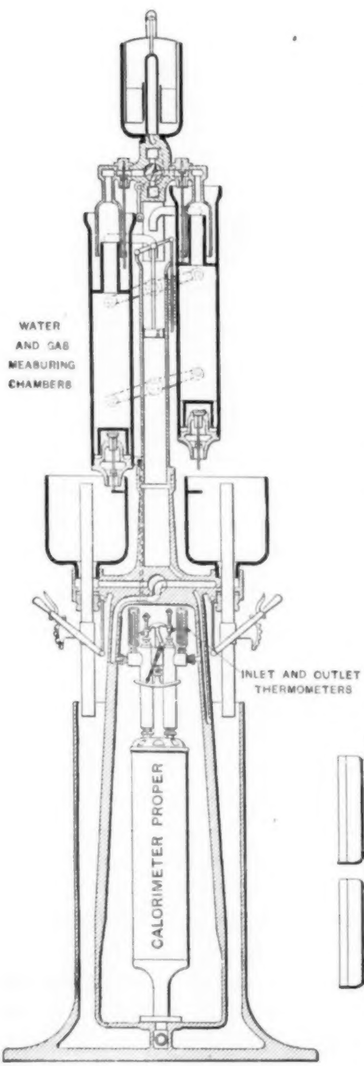


Fig. 3

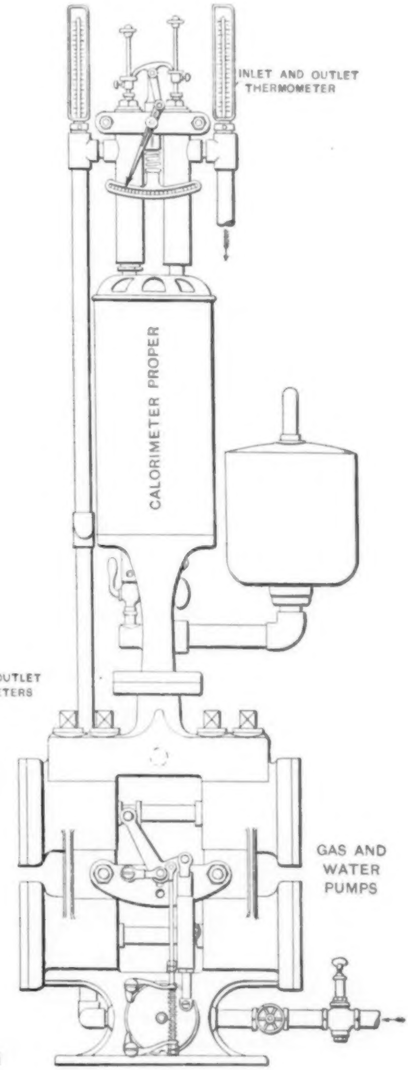


Fig. 4

Two Types of Calorimeter by Schutte and Koerting, with Means for Attaining Proportionality of Water and Gas

entering the instrument and into the water leaving the instrument two small liquid chambers are inserted, each provided with a small plunger projecting out through the top. As the liquid in this plunger chamber takes the temperature of the cooling liquid, the fixed body of liquid will expand and contract and the plunger projecting into it and held by a spring in one direction will rise and fall. The difference between the position of the two plungers with an initial setting indicates the difference between the temperature of the ingoing and outgoing liquid, with a certain time-lag of course, and

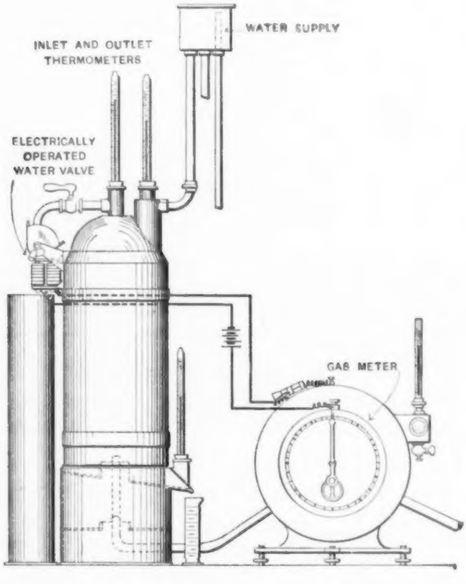


FIG. 5 SARGENT'S ARRANGEMENT OF THE JUNKERS CALORIMETER

is made to indicate on a dial by a piece of mechanism entirely mechanical.

21 One attempt to reduce the labor necessary to operate the ordinary Junkers instrument and also to eliminate the error of simultaneous reading of water and gas is shown in Fig. 5, which represents the instrument patented by C. E. Sargent, March 27, 1906. The new element in this case is the water valve controlling the discharge from the heating chamber, which is magnetically operated through electrical contacts on the gas meter dial. The instrument is however

intermittent in action and requires the attention of an operator to observe the quantities and to make calculations for calorific power.

22 An instrument of the automatic sort was patented by H. L. Doherty, August 14, 1906, which in principle is much the same as that described by Schutte and Koerting. The maintenance of proportions between the water and the gas is accomplished by Mr. Doherty by displacing the gas supplied to the burner from a tank, with the water that has passed through the heating chamber. Preparatory to using the gas it is allowed to remain in the tank long enough to acquire the temperature of the room and the final temperature of the g s is adjusted by varying the effective heating surface of the heater, thus reducing any error due to inequality of the temperature of the gas, air and fuel. The instrument is complicated, requiring some sixty figures in the patent specifications to illustrate it.

23 The various attempts to render the instrument automatic and continuous have, in the first instance, involved greater complication than the original instrument, whereas it is extremely desirable that less complication be introduced and real continuity of action, instantaneous equality of rates of gas and water flow, and maximum simplicity, be attained.

24 The method which I have proposed and used for accomplishing this is shown in Fig. 6 as applied to the heating chamber of the ordinary Junkers instrument. It is well known that a nozzle or fixed orifice, when supplied either by gas or water under a constant pressure, becomes a very accurate measure of the quantity flowing. In the instrument shown, water is allowed to flow to the heating chamber from a water nozzle, and gas to the Bunsen burner through a gas nozzle, both supplied from a constant pressure chamber. With these nozzles the constancy of ratio of the quantity of gas to quantity of water can be maintained without difficulty, the only possible sources of error being in a considerable change of pressure, which is improbable, or in a clogging of the instrument by dirt, which probably would not be serious by reason of the mode of introducing the gas by water aspiration and the ease with which nozzles can be removed.

25 At the top is shown a water aspirator which draws gas from the main through a water seal to remove the main pressure, discharging water and gas together into the chamber. The surplus water passes down the center through an overflow and the surplus gas goes through a water sealed by-pass shown here as a glass bottle. Gas is then taken from the top of the chamber to the Bunsen burner and water from the bottom of the chamber to the water orifice. The

water supply pipe to the heating chamber is arranged to bring the inlet water thermometer close to the outlet water thermometer and in the final form of such an instrument these two stems will be arranged so as almost to touch, permitting the use of a sliding scale, such as proposed by Junkers. There may also be used in place of these thermometers, and if desired in addition to them, a group of

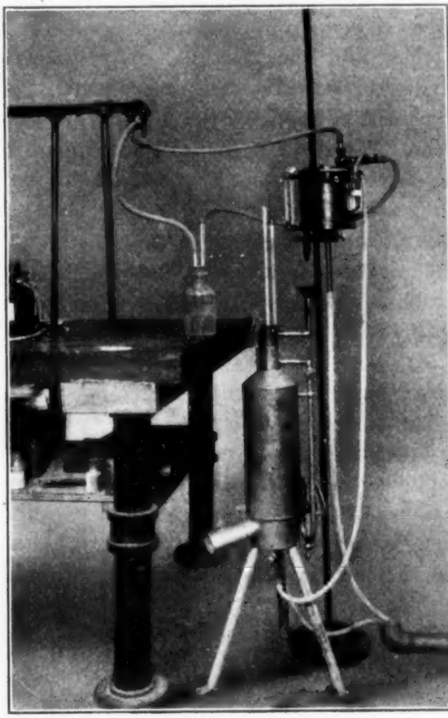


FIG. 6 JUNKERS CALORIMETER, ARRANGED BY THE AUTHOR FOR GIVING A CONTINUOUS RECORD OF THE QUALITY OF GAS

thermo-couples. At my request Prof. Wm. H. Bristol applied such a group of couples to this instrument in my laboratory at Columbia and recorded the temperature difference on one of his transparent smoke chart recording milli-voltmeters, which gives, so far as I know, the first continuous record made in this country of the quality of gas applied to a main for a considerable period of time.

26 It is evident, also, that the form of the chamber from which the water and gas are both supplied to the nozzle and the means of maintaining a constant pressure therein may be subject to quite a large number of variations in form to adapt it to the particular conditions or the taste of a designer, the requirements being simply that water and gas shall be supplied from a constant pressure chamber, and preferably supplied to that chamber; though not necessarily by a water aspirator, which would render both the gas and chamber pressure independent of pressure fluctuations of the gas mains. To adapt the instrument to different gases, it is necessary only to supply calibrated nozzles of different capacity. Thus, for rich natural gas a small gas nozzle and a large water nozzle would be used, while for producer gas or blast furnace gas a large gas and a small water nozzle. By the adjustment of nozzles any desired rise in temperature with any kind of gas can be secured and it takes only a minute or two to change the nozzles. Two chambers can be used if desired, one for water supply and the other for gas. This is especially desirable, even if not absolutely necessary, for gases containing matter that might be absorbed by water or lost by condensation, such as CO_2 or some hydrocarbons.

27 It is hoped that this presentation of the problem of continuous gas calorimetry, and one simple mode of solving that problem, will prove of some benefit to the gas power industry by removing one of the many uncertainties involving both the commercial and scientific phases of the question.

DISCUSSION

Mr. Henry L. Doherty I am the inventor of the calorimeter which Professor Lucke describes as very complicated, and if I am not mistaken, I am the earliest inventor of the displacement principle in this connection. When my application went into the patent office, no citations were given me of antedating patents.

2 The question of a continuous gas calorimeter is of considerable importance, especially for producer gas. The calorific value of a gas changes very quickly, and a continuous gas calorimeter that will not lag too far behind the operation of the producer is important, and one of the greatest considerations is to get a prompt record of the performance of the producer, and not have the calorimeter lag behind several minutes, as is apt to be the case. We have secured very good results by simply burning gas in an open flue, and recording the temperature of the escaping flue products. This is a scheme which is

positive and will give accurate results, but I have not applied for patents on this calorimeter, and have never put it on the market; nor have I ever put the other one on the market, which Professor Lucke terms complicated, and which, by the way, an inexperienced operator can run continuously within four B.t.u. on a gas of 600 B.t.u., and come within ½ of 1 per cent.

THE AUTHOR It is somewhat disappointing to note the small amount of discussion on this important topic of simple, continuous, indicating and recording calorimetry. An instrument for this purpose is just as necessary, as has been pointed out, to the proper management of gas engines and gas producers as the steam gage is in steam plant practice. The question of priority of invention in this line is one that cannot be settled in such a discussion; it is rather a question for the patent lawyers and the courts. The important point really at issue is that of the characteristics of structure and principles of operation which can be employed now in the solution of this important problem.

No. 1188

CLUTCHES

WITH SPECIAL REFERENCE TO AUTOMOBILE CLUTCHES

By Henry Souther, Hartford, Conn.

Member of the Society

POSITIVE CLUTCHES

The positive or jaw clutch is necessarily used only where the character of the starting action is immaterial, and if sudden, matters but little. Obviously it can be used only where the inertia of the standing or driven parts is relatively small, otherwise materials could not stand the wear and tear.

2 Modifications of the positive clutch are made in the angles of engagement between the jaws. The least positive form is one where the planes of engagement are inclined backward as regards the direction of motion at an angle of 15 deg., more or less. The tendency of such a clutch is to disengage under load. It must be held up to its work by an axial pressure, which may be regulated to perform a normal duty, but to slip and disengage when called upon abnormally by some accident or overload.

3 Positive clutches with engaging planes parallel to the axis of rotation must be held up to their work to guard against a natural tendency to jar out, but they present no safety features against an overload.

4 More positive yet in its action is the so-called undercut engagement of the jaw clutch, the tendency of which is to engage more tightly when loaded; and which can be disengaged only when absolutely free from load and in a condition to be rotated in a reverse direction enough to overcome the undercut angle.

5 In automobile construction the positive type of clutch is used inside the gear box, so arranged as to be operated only while the main

Presented at the monthly meeting (May 12, 1908) of The American Society of Mechanical Engineers. Discussion continued at the Detroit Meeting (June 1908).

friction clutch connecting the engine with the driven shaft is disengaged. Each gear carries a positive jaw clutch to be engaged with mates on the driving shaft (while the main friction clutch is open). Several inventors have accomplished the same object by means of a sliding spline (or hardened ball) on the driven shaft, which engages with the gear hub internally.

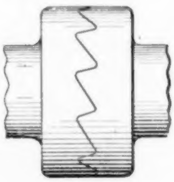


Fig. 1

Fig. 2

Fig. 3

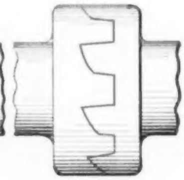


Fig. 4

6 The starting crank of an automobile is a first-rate illustration of an undercut positive clutch. It is undercut so that when the hand is applied to the starting crank there is little or no danger of the clutch slipping off and wrenching the operator. It is a fact, however, that

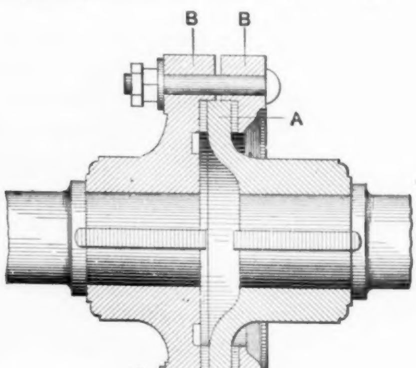


Fig. 5 Ramsbottom Clutch

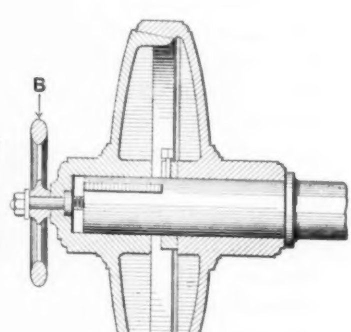


Fig. 6 Cone Clutch

some of these clutches are not undercut and are disagreeable to handle for this reason.

CLASSIFICATION OF FRICTION CLUTCHES

7 A rather careful search of the literature of clutches reveals the fact that there are a few basic types involved in all clutches, but

an infinite variety of detail in construction and manipulation. Rankine differentiates between friction clutches about as follows:

Friction clutch (contracting band).
Friction cones
Frictional sector (invented by Bodmer).
Friction disc (Weston's invention).

8 Reuleaux illustrates the Ramsbottom clutch as used for rolling mill work. This is nothing more or less than a friction coupling in which one flange is squeezed between frictional surfaces by being tightly bolted. Referring to Fig. 5, the flange attached to part *A* is firmly clamped beween the wood-lined surfaces of *B*, adjustment of the bolts being such that the friction will resist normal torque but yield to abnormal torque. This is perhaps the most simple form of friction clutch.

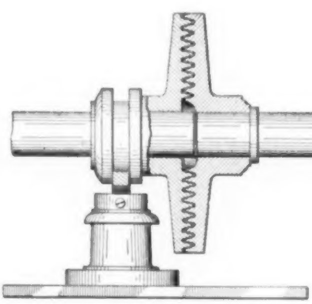


Fig. 7 Multi-cone Clutch

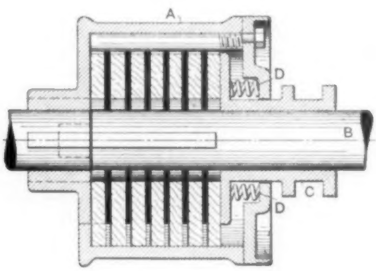


Fig. 8 Weston Clutch

9 Reuleaux then shows as the next step in the development of the clutch a cone coupling, the two parts being forced into engagement by screw and handwheel *B*, as shown in Fig. 6. He states that the angle of the cone should be not less than 10 deg., in order that the parts may not become wedged together. He also gives in connection with this clutch, with frictional surfaces of iron on iron, a coefficient of friction of 0.15. In order to keep the axial pressure within reasonable limits, he places the mean radius of the cone between three and six diameters of the shaft.

10 Following the single cone clutch in Reuleaux is what might be called a multi-cone, as shown in section by Fig. 7, a series of concentric cone-shaped rings with angles of 10 deg., or 20 deg. for both halves of the cone. As shown in this cut, it is apparent that the collar

would have to resist the pressure and wear due to the axial pressure necessary for proper engagement. This would be serious, and such wear is avoided in heavy machine work or in high-speed automobile design by making the axial pressure self-contained on the rotating member, except when the clutch is in the act of being disengaged. Such a construction is absolutely necessary in automobile clutches. This modification is shown in Fig. 9. The pressure of the screw wheel is self-contained, the two halves *A* and *B* being clamped together by it, the concentric double-faced cones furnishing much friction at slight axial pressure.

11 The next clutch shown by Reuleaux is that which he attributes to Koechlin, Fig. 10. This is of the internal-expanding type, three

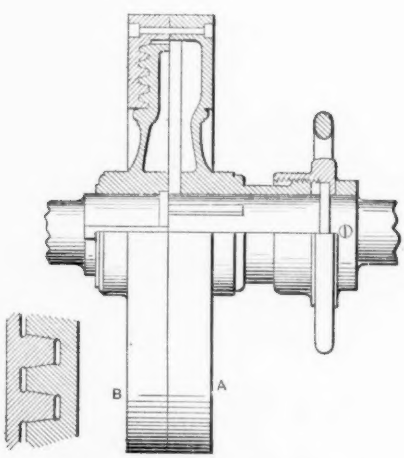


Fig. 9 Self-contained Thrust

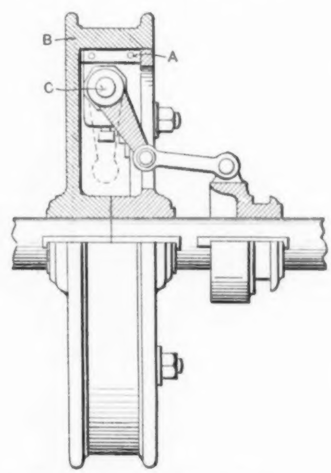


Fig. 10 Expanding Type

internal clamp pieces *A*, fitted with bronze shoes, being thrust out against the enclosing cylindrical drum *B*, by means of lever and screw action. Reuleaux points out that there is no danger of wedging in this clutch, such as exists in connection with the cone clutch.[1]

12 Reuleaux next shows a form of "axial friction coupling," well known as the Weston clutch, based on the principle of multiple-plate friction, Fig. 8. The plates are alternately wood and iron, as indicated, the wooden ones engaging with the outside cylindrical containing-case *A*, and the iron ones with the shaft *B*. In the form

[1] Bodmer and Koechlin seem to have been working along similar lines.

shown, the plates are pressed together by springs D, and released by drawing back a collar C, which releases the spring pressure.

MACHINE SHOP CLUTCHES

13 Perhaps the simplest form of machine shop clutch is that in which one flat disc presses against another, the surfaces being leather against iron, bronze against iron, or wood against iron, the axial pressure being great enough to drive the maximum load, yet not so great but that slipping takes place when the load is first applied, which prevents all jar. Such clutches are familiar in the driving of looms.

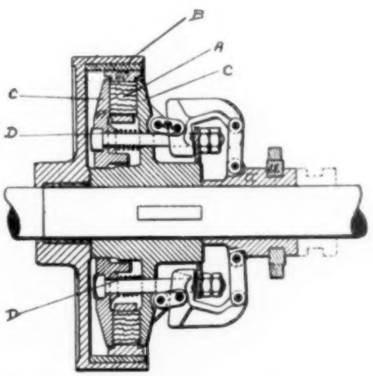


Fig. 11 Modification of Weston Type

14 In Fig. 11[1] is a modification of the Weston type. It is not multi-disc, there being only one wooden disc A, attached to the enclosing case B, which is gripped between two iron surfaces C, keyed to the driving shaft. To prevent any drag when disengaged, separating springs D are supplied, which part the frictional surfaces when idle. Slight rubbing when idle is not a very serious matter in machine shop clutches, however; but its importance in connection with automobile clutches, I will bring out later.

15 It is interesting to note that most information as to the frictional capacity of these machine shop clutches is empirical. The coefficient of friction of maple (which is commonly used) on cast iron is known. Little use can be made of this knowledge, however, as

[1] Fig. 11 is from the catalogue of T. B. Wood's Sons, Chambersburg, Penna.

the degree of lubrication, or the lack of it, may easily double or halve this coefficient. Manufacturers usually state in their catalogues the horse power that can be transmitted by their clutches at 100 r.p.m. It is probable that even such untechnical information is decidedly more reliable than that obtained from any formula containing an unknown variable—the coefficient of friction.

16 What was formerly known as the Frisby clutch was designed many years ago when no attention was paid to mathematical design, but its capacity has been well established by experience. Fig. 12 shows this clutch, which is not unlike the last one described, except that a flat surface *A* and cone *B* are used in combination. The gripping of the surfaces is accomplished in very much the same way. The frictional surfaces are separated by springs when disengaged.

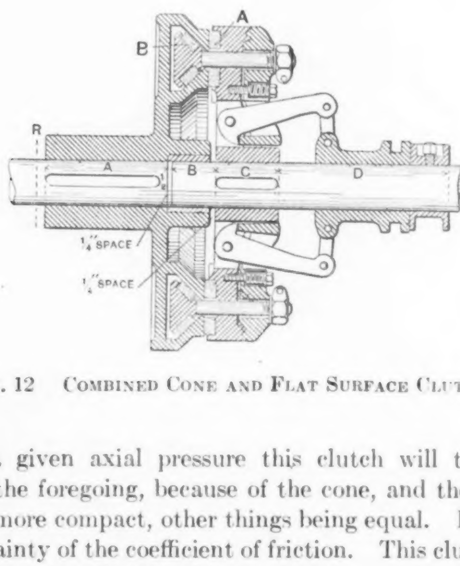


FIG. 12 COMBINED CONE AND FLAT SURFACE CLUTCH

17 For a given axial pressure this clutch will transmit more power than the foregoing, because of the cone, and therefore will be smaller and more compact, other things being equal. But here again is the uncertainty of the coefficient of friction. This clutch, for example, might throw its oil to the frictional surfaces more than the previous example, the oil more than offsetting the effect of the cone engagement.

18 Experimental data upon the capacity of the clutches of the Dodge Manufacturing Company, Mishawaka, Ind. are given in the following table. The results were obtained from clutches fitted with maple blocks and calculations are based on a coefficient of friction of 0.37 and a speed of rotation of 100 r.p.m.

Horse power	Block area	Diameter at block, inches	Circumferential pull at block center	Total pressure	Total pressure per square inch
25	120	16	1960	5300	44
32	141	18	2240	6000	44½
50	208	21½	2900	7800	37½
98	280	27½	4500	12 200	43½

19 A modern adaptation of the old Koechlin form of internal expanding clutch is shown Fig. 13 and 14 (from catalogue of the A. & F. Brown Company, Elizabethport, N. J.). These are largely used for very heavy work, the firm advertising clutches 48 in. in diameter capable of transmitting 320 h.p. at 100 r.p.m. The frictional surfaces are wood, especially prepared for the purpose, against iron; the experts of the company claiming that this combination is not liable to strike fire, as in cases where both friction surfaces are of iron

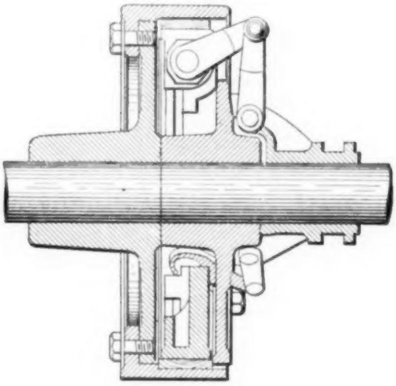


Fig. 13

Fig. 14

Modern Adaptation of Koechlin Clutch

CLUTCHES USED IN WIRE DRAWING

20 One of the oldest usages made of clutches is in the wire drawing art. The iron drum around which the wire is wrapped as a rule contains some form of clutch. What I believe to be the most recent development in this direction is that of the Morgan Construction Company, of Worcester, Mass. This is a compound

clutch, the main driving effort being furnished by a wrapping coil on a chilled iron surface, the initial engagement of the coil being brought about by a modified cone or ring slipping down onto a cone which drags the free end of the coil into engagement. Once seized, the wrapping continues until tight.

21 In Fig. 15, A is the tapered friction surface of the chilled drum on which the friction ring bears and below is the coil which is submerged in oil in an annular oil chamber. The drum is 12 in. diameter by 7 in. high and the coil, which is of soft steel, is $1\frac{1}{2}$ in. square at the large or driving end and $\frac{3}{8}$ in. by $\frac{5}{8}$ in. at the small end. The outside

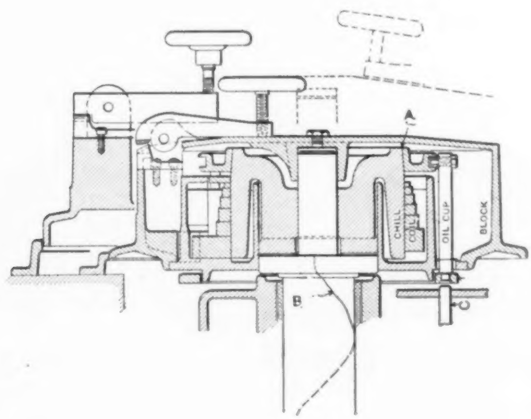


Fig. 15 Clutch for Wire Drawing

diameter of the block is 25 in. I quote from a communication received from the manufacturers of this clutch as to the importance of starting wire into a die gradually and smoothly.

Electrically operated wire-drawing machines in England have demonstrated that if wire can once be put into motion and the speed increased gradually to prevent breaking the wire, the possible speed of the drawing is almost unlimited. The wire upon which all the work is done becomes exceedingly hot, but the dies remain quite cool.

In order to test the power of our block, we keyed the hub to the shaft. This hub was flattened on one side. On this flat surface we strapped an 8 in. I-beam about 14 ft. long, and at a distance of 12 ft. 6 in. from the center of the driving shaft we strapped a gear to this I-beam in which we could put small weights. We first wrapped a $\frac{1}{2}$-in. annealed crane chain around the block, fasten-

ing one end to the block and one to the machine, with a weight of 500 lb. at a distance of 12 ft. 6 in. from the center of the driving shaft. The chain broke with a clear fracture. All the links of this chain were strained beyond the elastic limit.

We next took a $\frac{3}{4}$-in. chain and fastened it in the same manner and added weight up to 600 lb., including the weight of the beam, at a radius of 12 ft. 6 in. from the center of the shaft. At this point the cast iron quill, which had a bevel gear connected at one end and the chilled friction drum at the other end, ruptured, the crack extending from the top of the flange down the spindle a distance of 16 in. in a spiral of $1\frac{1}{2}$ revolutions, as shown at *B*, Fig. 15, the fracture showing clean, close-grained iron. This cast iron quill was 5 in in diameter and cast around a rough-turned shaft $2\frac{3}{16}$ in. in diameter. It had a flange at the top $8\frac{1}{4}$ in. in diameter by $1\frac{1}{4}$ in. thick.

This was the extent to which we carried out experiments and under the above conditions the friction clutch did not slip after it had taken hold.

The following calculation gives the pounds pull exerted on the chain:

$$\frac{600 \times 150 \times 49}{22 \times 12\frac{1}{2}} = 16\ 036 \text{ lb. (more or less)}$$

The pull exerted on the large end of the coil would be equal to

$$\frac{16\ 036 \times 12\frac{1}{2}}{6} = 32\ 761 \text{ lb. (more or less)}$$

The horse power of the clutch at 100 r. p. m. under the above conditions would be

$$\frac{32\ 761 \times 3.1416 \times 100}{33\ 000} = 302 \text{ h.p.}$$

22 A clutch of this kind has been in service some two years, drawing spring wire largely. No repairs or adjustments have been made during that time. The one commercial objection to it is its considerable cost, but it is expected that its good behavior will more than offset that.

CLUTCH OF SMALL DIMENSIONS

23 A strong demand has developed for a clutch of very small dimensions for a given capacity. This demand has been met in rather a curious way. Instead of cast iron or metal of ordinary strength, hardened tool steel frictional parts have been resorted to. This permits exceedingly high normal pressures between the frictional surfaces. Fig. 16 gives a very good idea of this form of clutch. It will be seen by inspection that the operating collar *A* forces wedge *B* between the long arms of the two levers *C*, spreading them in such a

manner as to expand a hardened steel ring *D* against the hardened steel enclosing drum *E*.

24 As much as 100 h.p. has been transmitted at 1000 r.p.m. with a clutch containing friction rings 5¼ in. in diameter and 1½ in. wide. This form of clutch has been largely introduced into automatic machines, machine shop countershafts and launch engines. Its engagement is apparently soft enough for any of these purposes, but in connection with automobile service it is yet in the experimental stage.

CLUTCHES WITH CORK FRICTION SURFACES

25 In connection with the commercial clutches of the forms now under discussion, cork has recently entered the field to a considerable extent and apparently with considerable success. It has a high coefficient of friction, probably double that of wood or leather on iron. Its behavior is peculiar because of its elasticity under compression.

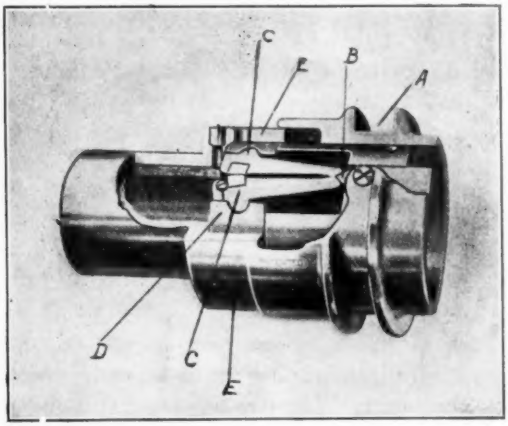


Fig. 16 Example of Clutch of Small Diameter

26 As a rule the corks are forced into suitable cavities formed for them in one of the metallic frictional surfaces. The corks are previously boiled and thereby softened and then pressed into the cavities. Thus established in a metal surface they normally protrude above the surrounding surface and engage first when the surfaces are brought together. If sufficient pressure is applied to the clutch they are

forced down flush with the metal surface and act with it in carrying the load. Following the release of the load they again protrude beyond the surrounding metal surface.

27 Two forms of cork are used, one in its natural condition, the other, which is stronger and firmer and yet very elastic, prepared as follows: small pieces are compressed into any desired shape under very great pressure and enough heat to cause the natural gums of the cork to exude and act as a binder. This form of prepared cork has not had wide use because of expense. Nevertheless, a clutch with cork friction surfaces will carry a greater load for its size than one of the usual materials. For example, a Dodge friction clutch carrying 500 h.p. gave much trouble on account of being overloaded. This clutch was strained up as tight as possible and it was all a man could do to throw it. The maple blocks used were replaced with compressed cork. It was then possible to loosen the adjustment of the clutch to such an extent that the operator could throw it with little effort. Following this change it was found that a set of cork blocks outlasted the maple ones five to one.

28 Prof. I. N. Hollis of Harvard University has determined the coefficient of friction of cork on cast iron to be from 0.33 to 0.37, the former for the heavier loads. For cast iron on cast iron he found the coefficient to be 0.16 and for bronze on cast iron 0.14. The coefficient of friction of cork on iron, therefore, apparently is about double that of iron on iron. It is claimed that the coefficient for cork is nearly as high when lubricated and that cork is much less affected by moisture than the maple blocks ordinarily used.

29 Other tests have been made by Prof. C. M. Allen of the Worcester Polytechnic Institute in connection with loom clutches. His results show a torque for cork inserts nearly double that of a leather-faced clutch for a given dimension.

AUTOMOBILE CLUTCHES

30 Soon after 1895 the evolution of the automobile or motor vehicle commenced in earnest. There was no difficulty in the way of operating the vehicle with steam or electricity. Positive connection between motor and wheels was quite possible because of the flexibility of the motor.

31 It was realized, however, that these were not the most desirable sources of power. The gas engine in its stationary forms was available. Starting as it does with an explosive impulse, direct con-

nection with the wheels of a vehicle was entirely out of the question. Consequently, a motor vehicle with a gas engine for prime mover was impossible without some means by which the motor and wheels could be separated during the starting of the motor.

32 In May 1879, Geo. B. Selden applied for a patent on a road engine in the United States Patent Office. His application incorporated the use of a clutch interposed between the engine and the gearing, so as to admit of running the engine while the carriage remained stationary. This is certainly one of the early recognitions of the importance of the clutch in automobile construction.

COMPARATIVE TESTS OF LOOM CLUTCHES

TORQUE MEASURED IN POUNDS FEET

Position of clutch	Pressure on clutch pounds	Torque	
		No. 1 "compo" clutch with cork inserts	No. 2 leather-faced clutch
Average results of eight positions	89.5	19.50	16.95
Average results of eight positions	151.5	34.20	17.66
Average results of eight positions	213.0	46.43	23.09
Average results of eight positions	275.0	57.05	29.46
Average results of eight positions	337.0	73.33	36.09
Average results of eight positions	398.0	82.24	41.31
Average results of eight positions	460.0	96.48	47.56

33 Perhaps the simplest form of automobile clutch is that commonly used for small machines and in connection with the planetary type of change gear. It operates by pressing one disc against another, the frictional surfaces being leather, bronze, or copper, against iron or steel. In Fig. 17 is shown a clutch of this type in use in a successful single cylinder automobile. The following data were furnished by the makers, the Cadillac Motor Car Company, of Detroit:

Maximum radius of leather frictional surface		$4\frac{11}{16}$ in.
Minimum radius of leather frictional surface		$3\frac{15}{16}$ in.
Mean radius of leather frictional surface		$4\frac{5}{16}$ in.
Area of leather frictional surface		$36\frac{1}{2}$ sq. in.
Axial pressure, from		1000 to 1200 lb.
Capacity	Horse power at 600 r.p.m	$5\frac{1}{2}$ h.p.
	Horse power at 1400 r.p.m	10 h.p.

34 The axial pressure in a clutch of this kind is usually produced by a spring disc; that is, the steel plate which carries the frictional surface, either leather or copper, is caused to operate like a diaphragm spring. The diameter for a car of 7 h.p. or 8 h.p. is usually from 5 in. to 10 in., the rubbing surfaces being from one-half to three-quarters of the entire superficial area of the disc. Such clutches are mostly available for two-speed cars, the disc clutch connecting with the engine direct and running at engine speed, the planetary system being used only for low speed and reverse work, actuated by contracting band clutches.

35 Motor vehicles so geared have their uses, but early in their development it was found that three or four speeds were desirable. Boxes

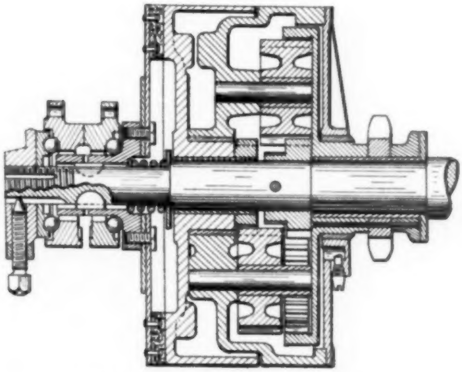


Fig. 17 Planetary Clutch

of sliding or change gears were resorted to, and here the character of the clutch became of prime importance. To be satisfactory, an automobile clutch used in this manner must engage and disengage easily, requiring but small axial movement of the operating mechanism, or of the clutch itself. It must be entirely independent of centrifugal force, and able to slip for a reasonable length of time without being destroyed.

36 The matter of absolute disengagement is perhaps the most important. Without it the sliding gears intended to be operated when the clutch is free or disengaged cannot be unmeshed nor remeshed. The slightest drag or friction in the clutch means a savage clashing of gears when changed. Gears with the teeth worn away were the rule rather than the exception in the early history of the art. This was, no doubt, due to the imperfect disengagement of the clutch.

37 The use of a system of gear change requiring the clashing of moving gears cannot receive the complete approval of the engineering world; yet this system has become a success by a combination of improvements in clutch and materials of which the gears are made, and treatment of the materials.

38 An important feature in a clutch is weight, especially as affecting its inertia. A clutch having high flywheel effect spins to such an extent as to cause violent clashing of idle gears. Consequently,

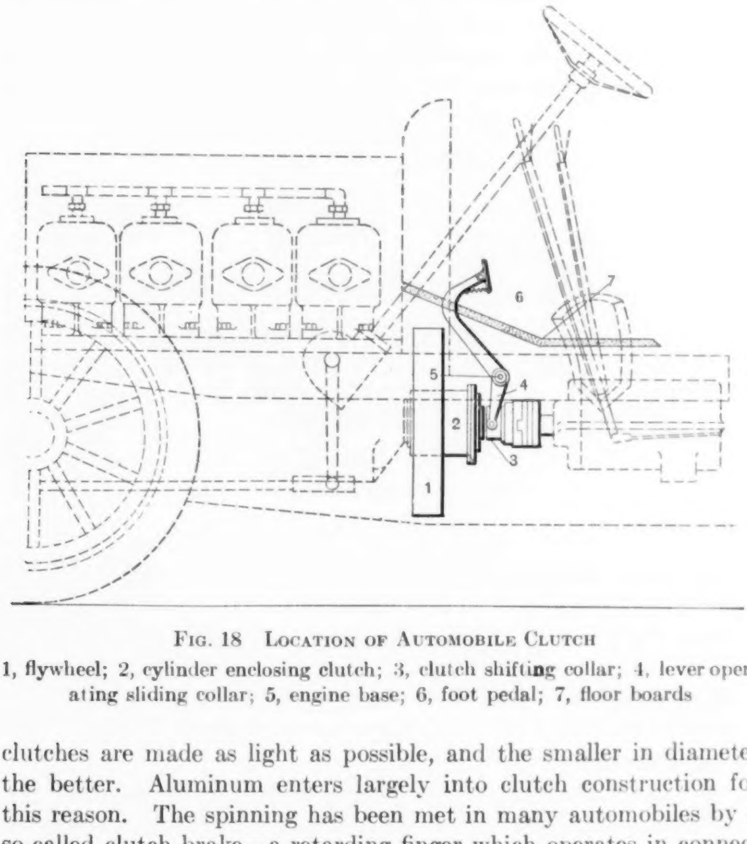


FIG. 18 LOCATION OF AUTOMOBILE CLUTCH

1, flywheel; 2, cylinder enclosing clutch; 3, clutch shifting collar; 4, lever operating sliding collar; 5, engine base; 6, foot pedal; 7, floor boards

clutches are made as light as possible, and the smaller in diameter the better. Aluminum enters largely into clutch construction for this reason. The spinning has been met in many automobiles by a so-called clutch brake—a retarding finger which operates in connection with the clutch-disengaging lever and bears upon some portion of the driven member of the clutch, braking it to a standstill.

39 Any automobile clutch must engage smoothly and absolutely without shock to be called a success. The quicker it seizes without

shock the better it is. Clutches exist that can be engaged suddenly and still not jar the passengers. But such a clutch is open to one very serious objection; that of not picking up the load quickly enough on a hill to start the car forward after a change of gears, before the momentum of the car is materially lessened. For example, in changing from the high gear to the next lower on a steep hill, a clutch that is too soft will permit the speed and momentum of the car to drop to such an extent that when the clutch finally does take hold the car is nearly at a standstill. This necessitates a further drop into a lower gear; one that will start a car from a standstill on a hill. The clutch designer is, therefore, between twc fires; too little slip on one hand and too much slip on the other. A degree of slip between the two must be found, and once found be capable of being maintained. It is doubtful if such a problem exists in connection with clutches anywhere else in the mechanical art.

40 The customary location for an automobile clutch is within the flywheel or at least at the rear end of the engine if the flywheel is at the front. Fig. 18 shows the application of the multiple-disc type with very little room between the gear box and the clutch, and only an Oldham coupling to give flexibility. This would be too close construction (as will be shown later) for the application of the cone type of clutch, which requires so much flexibility back of it; but the illustration gives a very clear idea of automobile clutch location in general.

THE CONE CLUTCH

41 I will take up the simplest form first, namely, the cone. I am pretty well satisfied that, all things considered, it is the best form when properly designed and mounted. It has the advantage of engaging and disengaging with very small axial motion. Axial pressure may be low because the normal pressure between frictional surfaces is multiplied by the angularity of the cone. Its weight may be very small, as the male member may be of aluminum. Its engagement is entirely independent of speed and centrifugal force. No liquid lubricant is needed with attending viscosity, drag, and change due to wear and temperature. Disengagement may, therefore, be made perfect.

42 Proper engagement, however, has proved a very difficult and baffling problem, and I think has caused nearly all the rejections of the cone clutch. A cone clutch may operate almost as savagely as a positive jaw clutch. It may also refuse to engage, if it does not

have the proper combination of angularity, pressure and lubrication. It may behave well at times and very badly at other times. A cone clutch of given angles and dimensions, with a definite axial pressure, may be a success in one car and an absolute failure in another.

43 The cause of this contradictory behavior may not be in the clutch proper; on the contrary, it is often in the surrounding mechanism of the clutch. The cone clutch must be absolutely free to center itself and seat itself uniformly. A short Oldham coupling or a single universal joint between the clutch and the driven shaft of the car is not enough to permit this under all conditions.

44 The weaving of the frame of the car puts cross strains on such a coupling, causing it to bind and causing the clutch to seize on one side before the other and be drawn suddenly into full contact. A change of angle, increased lubrication and a change of materials on the friction surface will not remove the trouble arising from this cause. A pair of generous, free-working universal joints must be provided, that the cone may reach its seat as intended.

45 Similarly, an engine mounted on a flexible sub-frame or pan support may move sufficiently to prevent the proper seating of the cone and cause a similar line of troubles. The male member must be mounted so as to be flexible enough to follow such small movements.

46 Experience has been a long time in teaching engineers that so much trouble can arise from apparently so small a cause; yet there are cases where misbehaving clutches have been made well-nigh perfect by the introduction of flexible couplings.

47 Leather (riveted to an aluminum cone) usually forms one of the rubbing surfaces and gray cast iron the other. The leather should be kept soft by neatsfoot or castor oil. Some builders boil the leather in tallow before applying to the clutch surface; but this matter is of minor importance compared with the mounting. With leather $\frac{1}{4}$ in. to $\frac{3}{8}$ in. thick, properly softened, engagement may be sufficiently mild, but an improvement is to place under the leather at six or eight points on the periphery of the cone flat or spiral springs that cause it to engage at these points a little bit before engaging elsewhere.

48 It is obvious that the construction surrounding the clutch must be such that an unusual supply of lubricant can by no means find its way to the frictional surfaces of the clutch. The flywheel prevents any oil from the engine working its way back, being provided with oil trap grooves for that purpose if necessary.

49 From the other direction, the gear-box, for example, oil

ordinarily does not get as far as the clutch, as there is usually a considerable space between. In general, it may be stated that the cone clutch is as free as any other from variations due to lubricants. The leather surfaces gradually become dry and hard, requiring the application of castor, or preferably neatsfoot oil, but not very often.

50 With proper usage, cone clutches with leather faces seem to last indefinitely. I have accurate knowledge of cars that have been driven surely 20 000 or 30 000 miles without replacement of the leather on the cone face, and I have yet to experience any trouble from the wearing of the leather face. A clutch that I had used for about 2500 miles gave no evidence of wear, and it had received no attention except a few doses of neatsfoot oil.

51 There is one defect in the operation of the cone clutch that has caused considerable trouble. The clutch necessarily requires some end or axial motion and a slip-joint that will permit it. An ordinary square slip-shaft has been commonly used. Instances have been found where these square slip-shafts have jammed under load and seized, so as to prevent the disengagement of the clutch at critical moments. Improved materials, increased dimensions and better facilities for lubrication have cured much of the trouble. Generous feathers and splines have been resorted to, which present working surfaces that are normal to each other and which avoid any cam-like or wedging action which may exist with a square shaft bearing in a reasonably easy-fitting square hole. Here, again, the perfect freedom introduced by double universal joints plays an important part, the square shaft being very much less apt to bind when perfectly free to center itself.

52 There has been a considerable variety of opinion as to the proper cone angle. Various authorities have placed it all the way from 7 deg. to 20 deg. The French have settled on an 8 deg. to 9 deg. angle as about right for a leather faced cone. Several important American makers are using 12 deg. to 13 deg., several 10 deg., and others 8 deg. The following table gives the dimensions of cone clutches used on three different models which are probably as successful as any:

Area of flywheel	113.1 sq. in.	78.7 sq. in.	73.59 sq. in.
Angle (one side)	8 deg.	8 deg.	8 deg.
Radius (maximum)	$8\frac{1}{2}$ in.	$8\frac{1}{4}$ in	$7\frac{5}{8}$ in.
Spring pressure	375 lb.	320 lb	250 lb.
Horse power by A.L.A.M. Formula	48	42	40

53 The metal-to-metal cone clutch may be made smaller in diameter and with a sharper angle, say 7 deg., without seizing, and may be copiously lubricated. This form has been used only to a small extent. The dividing line between slipping and seizing is narrow.

54 Another form of cone clutch has an aluminum male member of about 12 deg. angle bearing against cast iron and with cork inserts in the face of the male member. This clutch is not easily affected by a lubricant and, in fact, may be run with copious lubrication. This type has not been widely enough used yet to give sufficient knowledge as to the possibility of general application under many varying conditions.

55 Up to this time I have referred entirely to what may be called a direct-acting cone, one where the male part of the cone moves axially

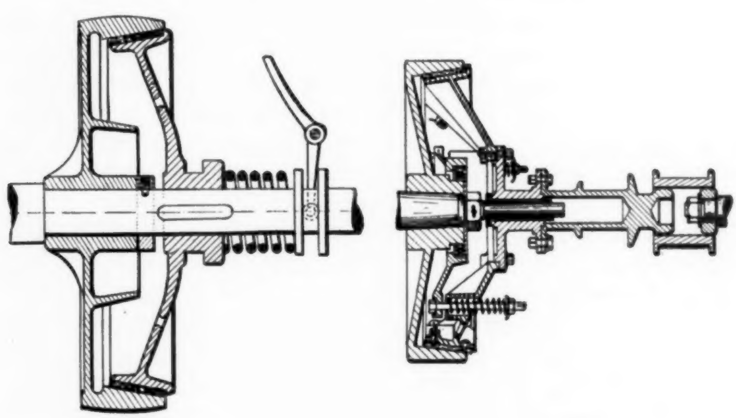


Fig. 19 Fig. 20

Leather-faced Cone Clutches

towards the engine, as illustrated in Fig. 19, which shows about the simplest form of leather-faced cone clutch. Modifications of this type are many. Fig. 20 shows a clutch of the same principle, but instead of one strong actuating spring surrounding its axis, it has three weaker spiral springs nearer the periphery of the male member.

56 Fig. 21 is a clutch used for a 50-h.p. car, with a cone angle of 13 deg., a diameter of about 16 in., a total frictional area of about 128 sq. in, and axial pressure of 375 lb. resulting from spring. This cut clearly shows a small spiral plunger spring A, underneath the leather face B, to make it pick up its load more quietly and smoothly. This

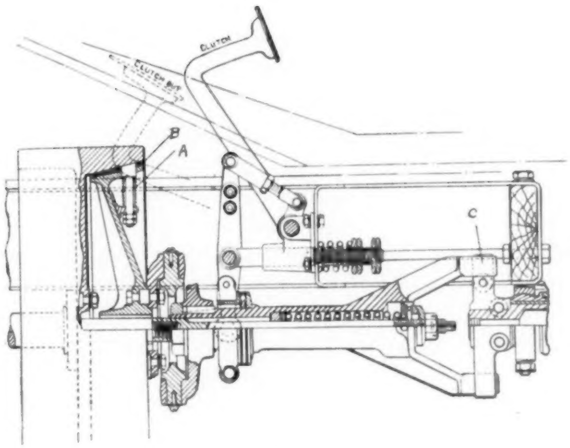


FIG. 21 CONE CLUTCH FOR 50-H.P. AUTOMOBILE

cut also shows a form of slip-joint back of the clutch C, which, although it does fairly good work, is not on the whole as satisfactory as the double-toggle universal joint. It will be noticed that the arms of this joint have been spread as widely as possible, but, at the best, the pressure and binding action is considerable.

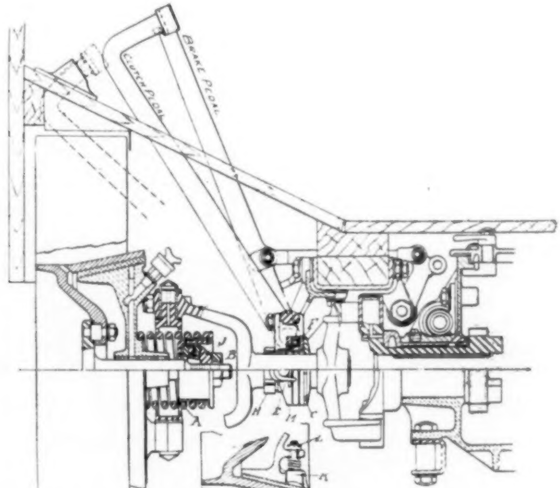


FIG. 22 CLUTCH OF SMALL DIMENSIONS FOR 30-H.P. CAR

57 In direct contrast to this clutch is the one shown in Fig. 22, where the diameter of the cone is very much less, not to exceed 10 in. This is a clutch used in connection with a car developing 30 h.p., A.L.A.M. rating, and one that has developed much higher horse power on the block—as high as 36 h.p. The clutch angle is 13 deg. and the frictional area the first two years this car was built was 86 sq. in., but this has recently been raised to 96 sq. in., the spring pressure remaining at 400 lb. It will be noted that at the bottom of this cut there is a sketch showing the spiral spring plungers underneath the leather.

58 Fig. 23 shows an early form of cone clutch used in 1902 or 1903 for a car of about 20 h.p. This has multi-springs for creating the proper frictional contact and a peculiar form of spring application,

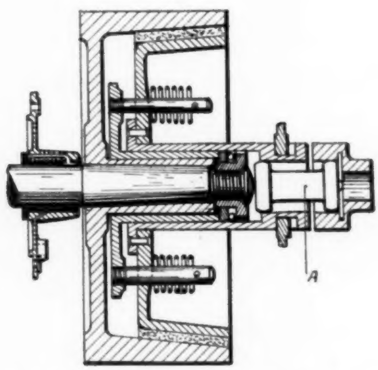


Fig. 23 Early Form of Cone Clutch

simple in the extreme. One of the early forms of toggle joint is also shown at *A*. This gave in its day what was considered very good service.

MODIFICATIONS OF CONE CLUTCH

59 The so-called inverted cone is well illustrated in Fig. 24. The reversed cone is contained in an extension *A*, built on the flywheel *B*. When the cone is disengaged it moves toward the engine, exactly reversing the action of the foregoing type. This clutch has its adherents, and it is a good one, differing very slightly in efficiency, if properly assembled, from the direct acting cone. It may be kept free from dirt and oil much more easily than the other form.

60 The following is a simple formula for calculating the ordinary

cone clutch, contributed by Chas. H. Schabinger to The Horseless Age of October 2, 1907:

$$h.p. = \frac{P f r R}{63\,000 \sin \theta}$$

P = Assumed pressure of engaging spring in pounds.
f = Coefficient of friction, which in ordinary practice is about 0.25.
r = Mean radius of the cone in inches.
R = Revolutions of the motor per minute.
$\sin \theta$ = Sine of the angle of the clutch.

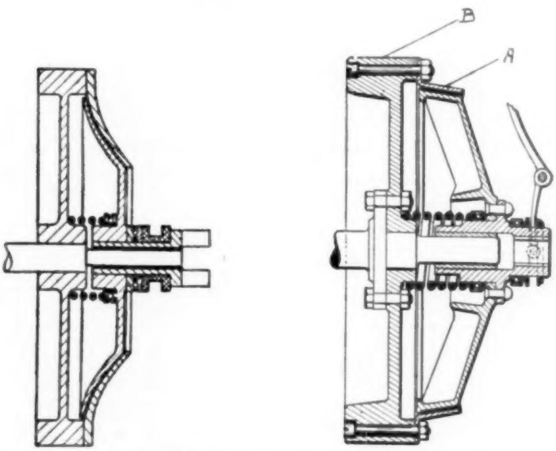


FIG. 24 MODIFICATIONS OF CONE CLUTCH

61 To obtain the size of spring when the horse power is known, the following formula may be used with good results:

$$P = \frac{h.p.\ 63\,000 \sin \theta}{f\ r\ R}$$

the same symbols being used as in the preceding formula.

62 It will be noted that the coefficient of friction used is 0.25. This is probably near enough for a properly lubricated leather-iron clutch.

EXPANDING BAND CLUTCH

63 The next type of clutch may be classified as internal expanding band or ring. This has had many exponents in the automobile

art, but is open to centrifugal effects to such an extent that it requires considerable ingenuity to overcome troubles arising therefrom. At

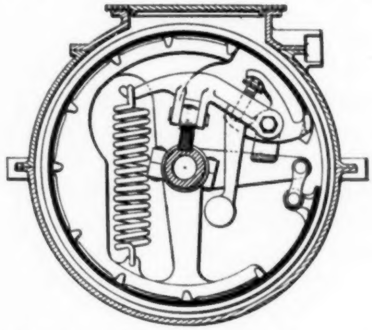


FIG. 25 EXPANDING BAND CLUTCH

high engine speeds the operating levers have in many cases been so arranged as to lower the normal pressure between the frictional surfaces, resulting in a slippage and arbitrarily fixing a maximum limit

FIG. 26

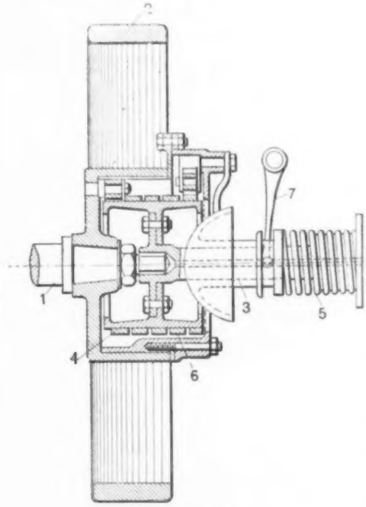


FIG. 27

MODIFICATIONS OF BAND CLUTCHES

of speed for the car on the high gear, and of horse power possible to develop in low gear.

64 Fig. 25 shows a clutch operating on the same principle, driv-

ing, a 16-h.p. car, the spring pull being 50 lb., the diameter of the clutch about 9.50 in., and the width of the band 2 in.

65 This clutch was particularly soft in operation, but did release at high engine speeds. It operated best with a certain definite quality and quantity of lubricant, which, if varied to any great extent, caused a slipping clutch or a sharply biting clutch. The tendency of the clutch is to unwrap and expand against the enclosing cylinder as soon as any friction is applied to it.

66 The successor of this clutch, shown in Fig. 26, was designed to overcome the centrifugal releasing effect of levers in the clutch

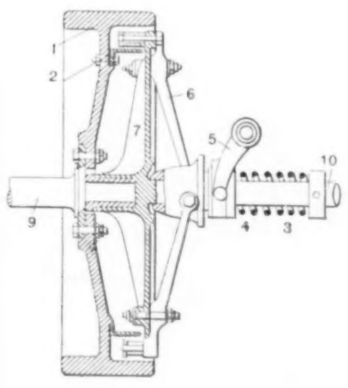


Fig. 28

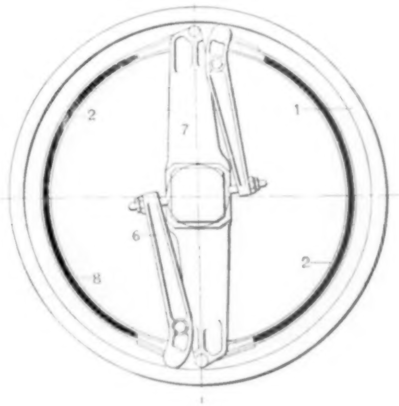


Fig. 29

Contracting Band Clutches

shown in Fig. 25. The total area of the clutch is 36 sq. in. and the two springs are each of 125 lb. tension. This clutch was a success, but was finally given up in favor of a simple cone.

CONTRACTING BAND CLUTCHES

67 The exponents of the contracting-band type of clutch are few and far between, unless the contracting spiral, Fig. 27, be so classed, as perhaps it should be. Fig. 28 and 29 show a contracting band, characteristic of one of the prominent French cars (Mors). A leather-lined flexible steel band (*8*) contracts against a steel cylindrical band (*2*) bolted to the flywheel (*1*). Clutches of this character are seldom found in the automobile industry, except in two-speed cars.

68 About 1897 a single cylinder, 10-h.p. car was equipped with a clutch such as shown in Fig. 29, a leather-lined band, very flexible

in character, wrapping around the hub of a flywheel and tightened with a spring pressure of about 50 lb. against a wedge. In this clutch a weight was furnished which would throw out at high speeds and further tend to tighten the band about the hub of the flywheel. The fact that this clutch has not had any successors is an indication that it could not compete with other forms; nevertheless, it was a successful clutch, especially for its time.

69 Fig. 27 shows a form of clutch that has had prominent adherents. It is the wrapping spiral spring, of either hard or soft metal. The cut indicates the spring in cross section, marked *6*, wrapping on the drum *4*. Probably the greatest enemy to this clutch has been the adjustment of the clutching force. With too little lubrication

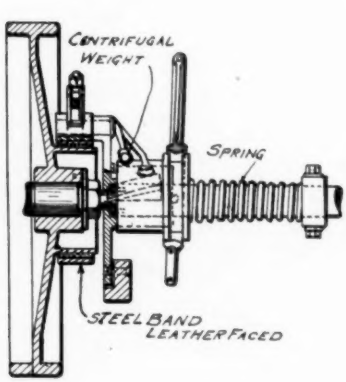


FIG. 30 WRAPPING BAND

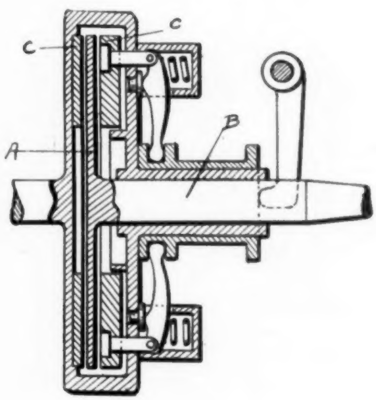


FIG. 31 SINGLE DISC

these clutches grip too savagely. With too much lubrication they will not pick up their load rapidly enough. The margin is narrow and hard to control. With a viscous lubricant there is enough drag to make the gears clash badly. The disengagement is not very complete at the best.

DISC CLUTCHES

70 The single-disc clutch is widely used, both here and abroad. It is so characteristic of a French make as to take the name of the firm—the De Dion. It is now used in this country by one firm for horse powers ranging from 70 to 20, for pleasure and for commercial service. The clutch has a disc *A*, Fig. 31, on the driven member

B, which is clamped between two discs *C*, on the driving member or flywheel. In Fig. 31 this arrangement is clearly shown. There are the necessary accompaniments of separating springs, so as to make disengagement perfect, also either single or multiple springs to cause the proper engagement.

71 Fig. 32 shows the same kind of a clutch in a slightly different form. The springs in this case are on the front side of the flywheel rather than on the rear, as in Fig. 31. Cork inserts are being used in this clutch to considerable advantage.

72 Another form is the now popular multi-disc clutch; that is, the elaboration of the Weston clutch, to which I have already referred.

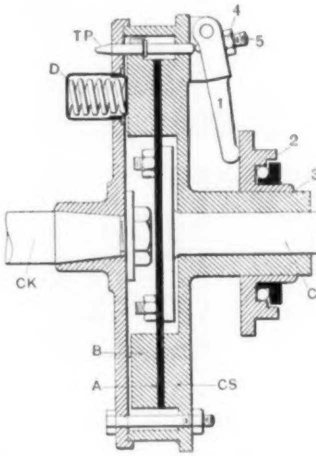


Fig. 32

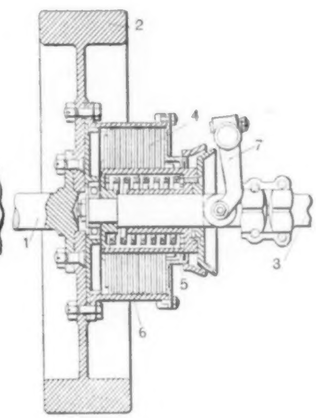


Fig. 33

Types of Disc Clutches

This clutch is indicated clearly by Fig. 33, the alternate plates of bronze and steel attached to driving and driven parts being pressed together by a powerful spring, *5*.

73 The question of lubrication is the all-controlling one here, in fact, the principal problem in connection with the multi-disc clutch as a type seems to be proper lubrication. I have ridden in cars equipped with such clutches that were extremely savage in taking hold, and in others of the same make that were extremely slow. In a way, this is desirable. For example, in a hill-climbing contest or race one would wish to pick up quickly and would be perfectly willing to put up with a harsh clutch and lubricate accordingly. On the

other hand, a car running about a level city, encountering few bad hills, would be able to lubricate excessively and still have a satisfactorily driven automobile. A clutch so lubricated would be extremely soft, and yet pick the car up fast enough for ordinary purposes on level roads.

74 Cold and heat affect the operation of this clutch, the lubricant in summer being thicker than can be permitted in winter. As it runs in oil it takes a certain length of time for the oil to squeeze out when engaged and for the metal to come in contact with metal and

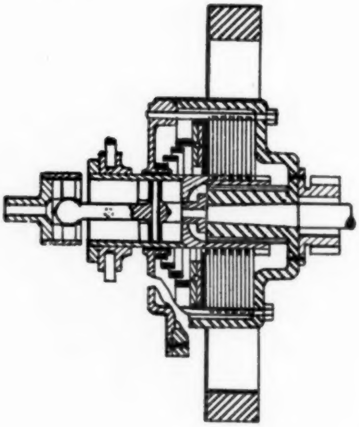


Fig. 34

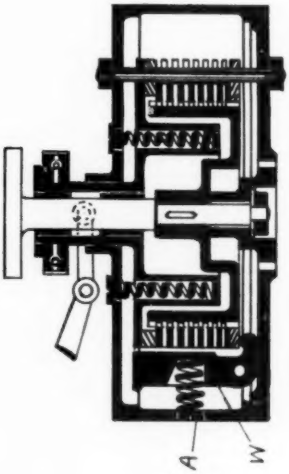


Fig. 35

Types of Multi-disc Clutches

really begin to drive the car. It will be seen from this that the viscosity of the lubricant is of prime importance.

75 One form of multiple disc clutch in use in a very high grade car consists of steel discs rubbing against a special bronze rolled into sheets. The steel discs are provided with several small tongues on the outer periphery, each bent one side enough to come in contact with the next steel disc, for the purpose of separating the discs and overcoming the drag when the clutch is disengaged. A small clutch brake is also provided to overcome the inertia or drag inherent in the clutch and due to viscosity of lubricant. The steel discs are put into the clutch as received from the rolling mill, with the hard black finish characteristic of carefully finished crucible sheet steel.

76 This clutch is connected with the crank case so that oil feeds into it from the crank case through a hole drilled in the center of the crank shaft. Entire reliance, however, is not placed on this supply, a little extra oil being supplied every two or three days through holes provided for that purpose.

77 Another form of this same type of clutch is shown in Fig. 34. Comparatively few discs are used as will be noted. On the other hand, it is apparent that the spring pressure is very heavy. This is a successful and well behaving clutch used on a popular car at the present time. It drags but little when the gears are changed and is satisfactory in that respect.

78 A type of disc clutch consisting of all-steel discs with alternate ones faced with leather was operated without any oil whatsoever, the leather being softened and made more or less pliable like the leather on the simple cone clutch. These clutches gave some trouble by burning up, the slip required to start smoothly being also sufficient to create enough heat to destroy the leather. This clutch was, however, extremely efficient in the transmission of power. For example, one with discs about 7 in. in diameter was powerful enough to drive an automobile of 50 h.p.

79 It must be remembered that the automobile engine runs at high speed, say 1000 to 1200 r.p.m., when developing anywhere near its normal rating, some motors, in fact, running up as high as 1500 to 1800 r.p.m. (standard rating is at 1000 ft. per minute piston speed).

80 It is a fact that in service cars with disc clutches of this character vary more or less in the way their clutches behave. Clutches receive very much less attention than they ought, like everything else on the automobile. I think it will be admitted, even by the adherents of this form of clutch, that it ought to receive more attention than the leather-faced cone. Nevertheless, this is now a very successful type of clutch, largely used in many high grade cars.

81 In the matter of the number of plates in the disc clutch there is no agreement between designers. Some use a very large number of thin plates, as many as 50 or 60, and others use a very small number, as few as six or eight; in fact, it may be said that the single disc clutch, which has only two frictional surfaces, is the lower limit.

82 In one very ingenious design for the multiple disc clutch, made by the Sturtevant Mill Company, of Boston, the pressure on the discs is brought about by centrifugal force acting on weights so arranged as to press the tighter with increased velocity. This is shown in Fig. 35. One of the weights is at *W*. It will be noticed

that this weight operates against a spring A, which prevents its flying out and gripping at too low an engine speed. Once, however, this spring pressure is overcome, the discs indicated by the alternately light and dark spaces are pressed together.

83 It would seem that this principle has one serious defect in the fact that at low engine speeds the gripping tendency would be small. It would, therefore, not be possible to develop high torque at low speeds, which is sometimes quite desirable. It is a fact, however, that it is almost impossible to stall an engine by applying this clutch too quickly; it does its own releasing so promptly and automatically.

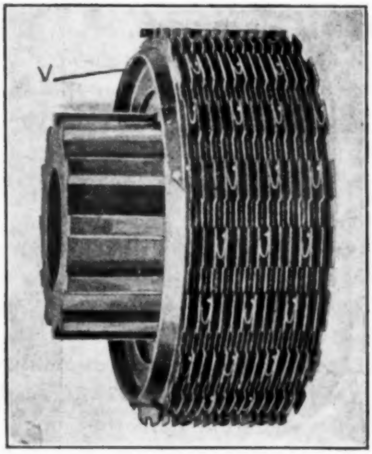


FIG. 36 MULTI-DISC CLUTCH

84 This principle has been elaborated in connection with an automatic change of gears: gear No. 1 being picked up at a given rate of revolutions by its set of disc clutches; gear No. 2 by an increased number of revolutions by a separate set of discs, and so on. In driving a car so equipped the changes take place without being perceptible except with the closest observation.

85 This system is open to the objection, however, of not being able to spin the engine very rapidly and connect with the low gear, in order to jump the car out of a hole or some unusual situation. I understand this has been overcome by supplying an independently operated lever for the foot, to be used in emergencies only.

86 With the automatic Sturtevant multi-disc clutch it has been found experimentally that, for the maximum slip speed usual in automobiles, 15 to 20 lb. per square inch pressure is safe, and that lubricated cast iron discs scarcely wear out the tool marks after many thousands of miles use. The makers state that experience has shown that safe slip is merely a matter of good lub ication and low pressures. They have experimented with small cast-iron discs, running dry and with constant slip at 2 lb. per square inch pressure, and even at that the clutch wore many weeks transmitting a heavy load.

87 A modification of the multi-disc clutch in which the cone and the disc are combined is attracting much attention. This clutch

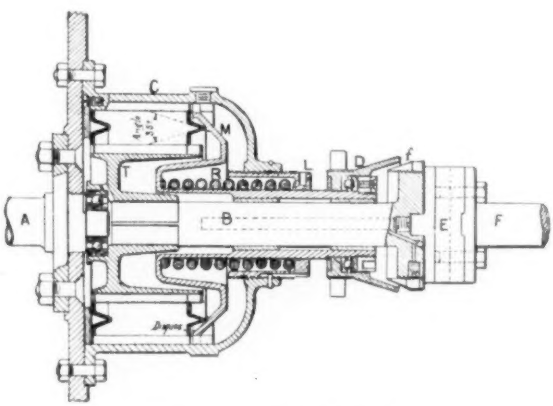


FIG. 37 V-SHAPED DISCS

(Hele-Shaw) is fully described by its inventor in the Transactions of the Institute of Mechanical Engineers (Great Britain), July 1903. Fig. 36 shows a set, or "pack" of discs from such a clutch. The U-shaped separating rings which force the discs apart when the spring pressure is released are plainly visible. This action overcomes the natural tendency of the oil to cause the discs to adhere. Each disc has a concentric V-shaped ring struck up in its surface, clearly shown in the end disc in Fig. 36.

88 Fig. 37 also shows the V shape of the discs very well indeed; in fact, the whole clutch is well shown here in section. Only the *V* portions engage, the surfaces of the discs not bearing. When the clutch is thrown out, the cone *D* bears on the cone *f*, thereby checking the spinning tendency of this clutch, or, if the viscosity of the oil is

heavy, holding it quiet during the changing of gears. This clutch is copiously lubricated and the V or engaging portions of the discs are perforated with holes so that the oil may circulate quickly in and out of the V grooves as they are engaged and disengaged. Outside the V portions of these plates or discs there is a comparatively large space between them, permitting the free circulation of oil and consequent rapid carrying away of heat if the clutch slips much.

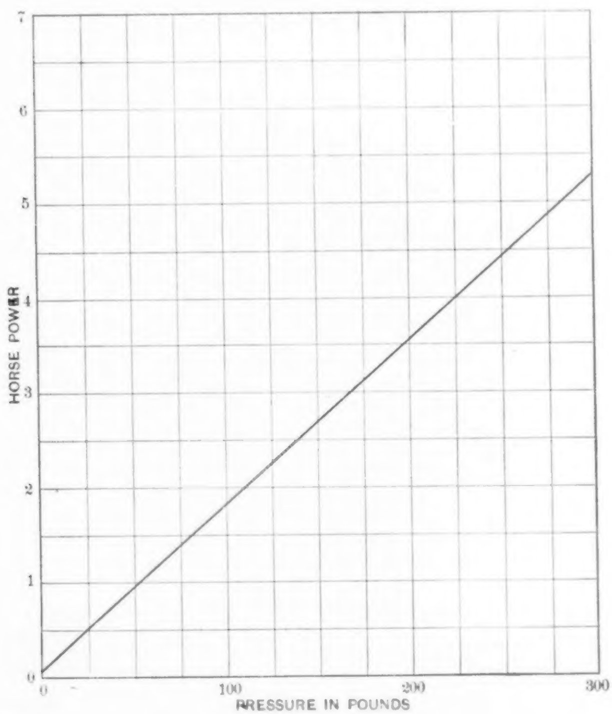


FIG. 38 POWER CHART FOR HELE-SHAW CLUTCH

89 In connection with the article referred to, in the Transactions of the Institute of Mechanical Engineers, are data on power transmitted by various spring pressures, given in Fig. 38. Fig. 39 shows the character of the curve depending upon horse power and pressure of springs.

90 One-thousand horse power is being transmitted by one of these clutches running at 700 or 800 r.p.m. and measuring 18 in. in

diameter between the V's in the discs. The following table gives the dimensions and number of plates used for different horse powers:

	Bronze	Steel
25 h.p., 27 plates of 6½ in.	14 outer	13 inner
40 h.p., 25 plates of 8½ in.	13 outer	12 inner
60 h.p., 21 plates of 11 in.	11 outer	10 inner

The space in length required inside of the clutch casing for 25 plates is 5 in., this including the space for the disengaging movement and the spring pressure plate.

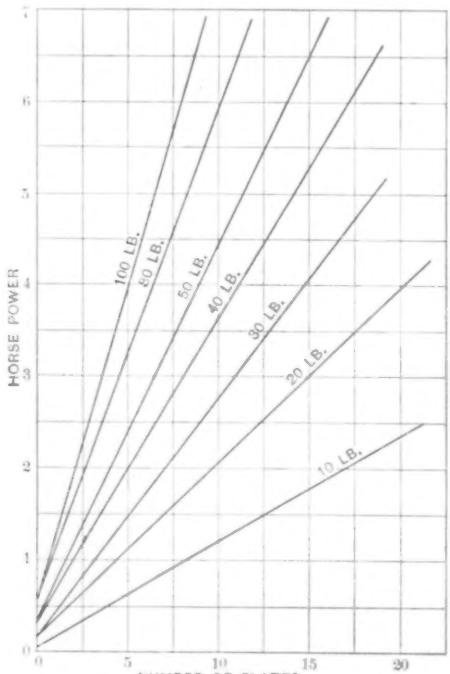


FIG. 39 CHART OF SPRING PRESSURES AND HORSE POWER

91 The number of plates in this clutch is made to vary with the power transmitted, the diameters remaining the same within certain limits. The principle involved is that the thickness of the pack of plates shall not exceed the diameter of the plate. When this becomes necessary in order to transmit a load, the plates are increased in diameter, fewer of them being used. The clutch is necessarily

heavy, but this is partially offset by the relatively small diameter. It has, consequently, little spinning tendency.

92 The materials for disc clutches in general have been various; namely, steel on steel, steel on leather-faced discs, steel on bronze and steel discs with cork inserts.

93 I have recently been informed of a disc clutch with cork inserts of natural cork that wore out twice in succession in about 1000 miles. This same clutch was equipped with compressed cork inserts previously described, which have driven the car some 5000 or 6000 miles without perceptible wear.

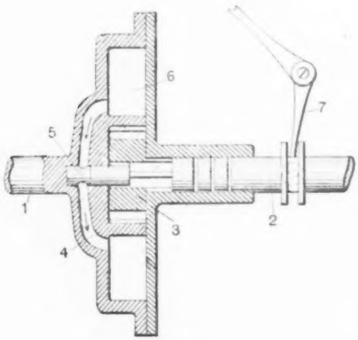


Fig. 40

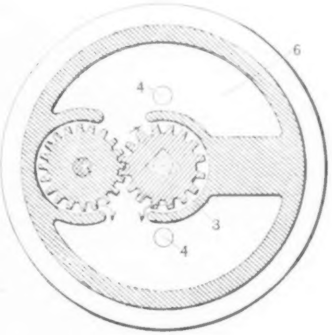


Fig. 41

Hydraulic Clutch

94 It is a fact that steel discs against steel have become badly heated and cut to such an extent as to make the clutches inoperative. Steel against bronze, however, does not seem to cut in this manner and the wear after two years' steady use is only 0.002 in. or 0.003 in. at the outside edge of the discs. I have not heard of the original combination of Weston, that is, wood against iron or steel, being used in connection with automobiles.

95 A pneumatic clutch has been developed by the Northern Motor Car Company, but has not been extensively used because of its cost. It has a plain leather-faced disc pressing against a metal plate, and air from a small pump deflects the leather, causing it to bear against the disc. Hydraulic clutches have also been used, but are not popular. Fig. 40 and 41 show one of the simpler forms.

96 The magnetic clutch is in use and is fairly successful. Such clutches are operated on the same principle as the so-called "pick-up magnet" found in so many plants. One complication arises in the

fact that one of the parts of the magnet has to rotate continuously, the gears being always in mesh; consequently the exciting current has to be carried to it by a brush. These clutches do not heat badly, at least not badly enough to cause any trouble; but they seize rather savagely unless carefully controlled. A considerable current is also necessary on the car for the operation of the clutch. These complications rather interfere with extended usage.

DISCUSSION

Mr. C. R. Gabriel The following figures show a few of the oldest and best known clutches as applied to power presses, many of which are in use today and preferred by some users to more recent designs.

2 Fig. 1 shows a type of press that has been in use for about sixty years, known as the Fowler press. The feature of the clutch in this press is that the driving pins are carried in the hub of the flywheel and operated by a sleeve on the outer end of the shaft, the pins engaging an abutment on the shaft collar. This arrangement permits of starting or stopping the shaft at any point of the revolution, and is called for by some users at the present time.

3 Fig. 2 shows the original Stiles clutch. This clutch has a spring actuated bolt and retaining latch carried in the hub of the fly wheel, the bolt latch being released by the action of the cam carried in the bracket below the shaft collar, which is connected to the foot treadle rod. Depressing the treadle allows the bolt to engage the recess in the shaft collar; a revolution of the shaft follows, and is automatically stopped if the foot has been removed from the treadle. Continued pressure on the treadle results in continuous strokes of the press. This clutch also permits of the shaft being rotated by means of a crank applied to the shaft on the connection end, without stopping the fly wheel, a very convenient feature in setting tools. Also more than one bolt may be carried by the fly wheel, which results in a marked saving of time on work when the press is stopped at each stroke, as the arc of rotation of the wheel will be in proportion to the number of bolts used, and the loss of time after the treadle is depressed will be diminished by the same ratio.

4 Fig. 3 shows a safety latch applied to the Stiles clutch, and designed to prevent repeating. If an operator holds his foot on the treadle for an interval of time greater than is consumed in one revolution of the shaft, the press will repeat, and many accidents are due to

this cause, as the operator attempts to remove or place the work and is caught before the hand can be removed from the dies. This arrange-

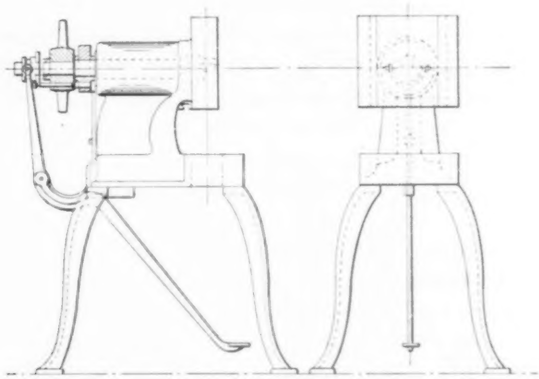


FIG. 1 FOWLER PRESS

ment consists of a type of releasing mechanism. A cam is secured to the shaft collar, and as the shaft rotates, it engages the toe of the hook lever, and forces it into the position indicated by the dotted lines,

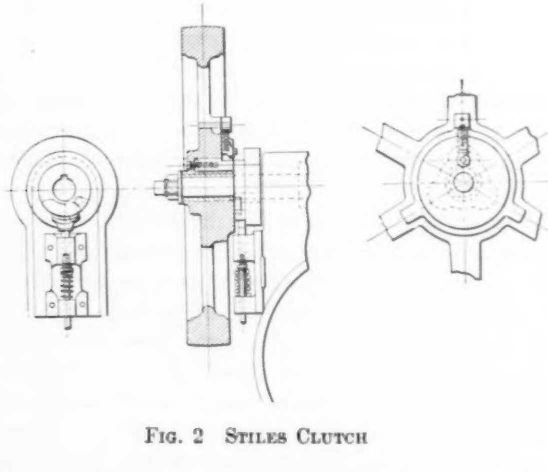


FIG. 2 STILES CLUTCH

thus releasing the clutch cam which is forced upward by the spring and disengages the bolt, stopping the press even though the operator keeps the treadle depressed.

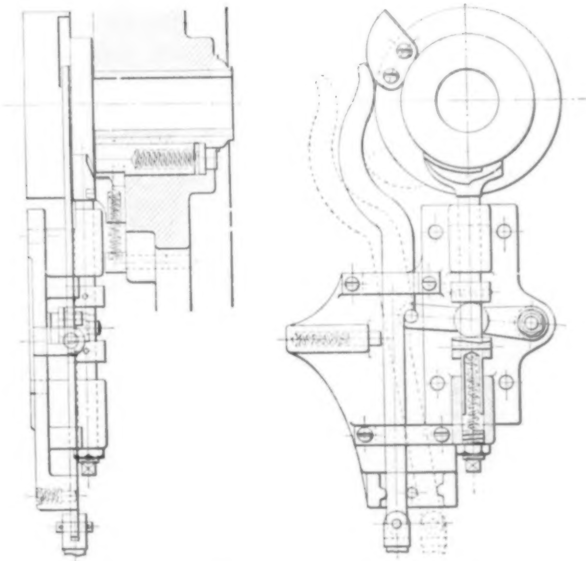


FIG. 3 SAFETY LATCH ON STILES CLUTCH

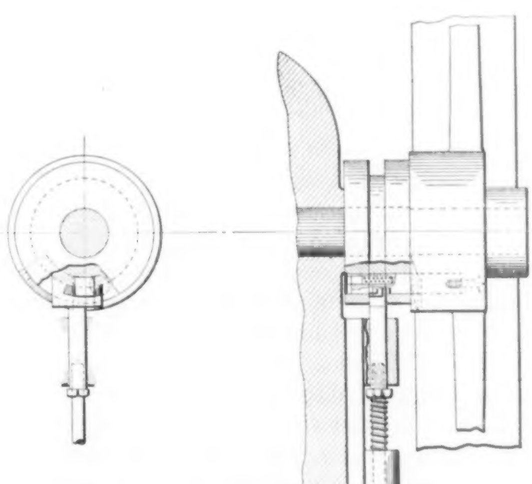


FIG 4 JOHNSON CLUTCH

5 For continuous running, the hook lever may be shifted to the right as indicated by the dotted lines at the bottom of the bracket; this carries the toe of the lever out of the path of the cam, allowing the hook to remain engaged, and continuous running is obtained by keeping the treadle depressed.

6 Fig. 4 shows the well known Johnson clutch, in very general use today. It differs from the types that have been shown by having the sliding bolt in the shaft. This construction does not permit of the shaft being rotated independent of the wheel, as the bolt is free

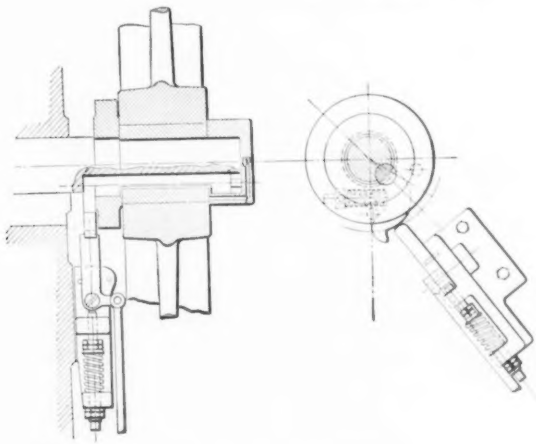


FIG. 5 BLISS CLUTCH

to engage the wheel as soon as it disengages the pull-out cam, which operates when the slide is at the up-stroke.

7 In setting tools it is necessary to remove the belt from the fly wheel and allow the clutch to engage and rotate the shaft by turning the wheel by hand. To save time when the press is to be stopped at each stroke, two or more notches may be made in the wheel hub.

8 Fig. 5 shows the well known Bliss clutch, which has the rolling pin instead of the sliding bolt. This pin is made of tool steel and engages a tool steel striking piece the entire length of the wheel-hub. As the clutch pin is located very close to the axis of the shaft, the velocity and violence of the blow with which the wheel is driven against the pin is reduced to a minimum. This assures the smoothness and durability that distinguish this type of clutch.

9 It has an advantage in that the wheel can be backed up with-

out rotating the shaft, which permits of bumping the shaft over the center, as is often necessary when the dies become jammed or in try-

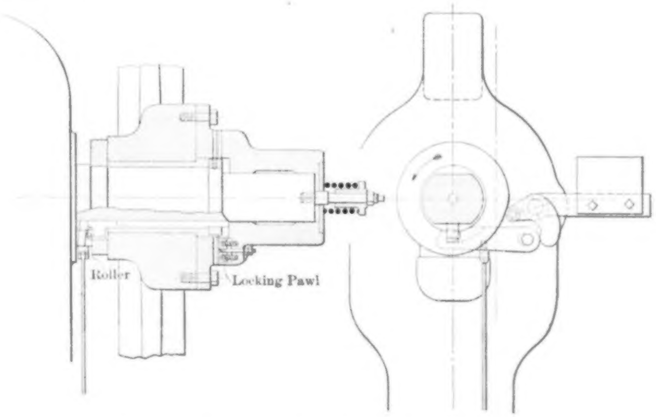


Fig. 6 Positive Multiple Jawed Clutch

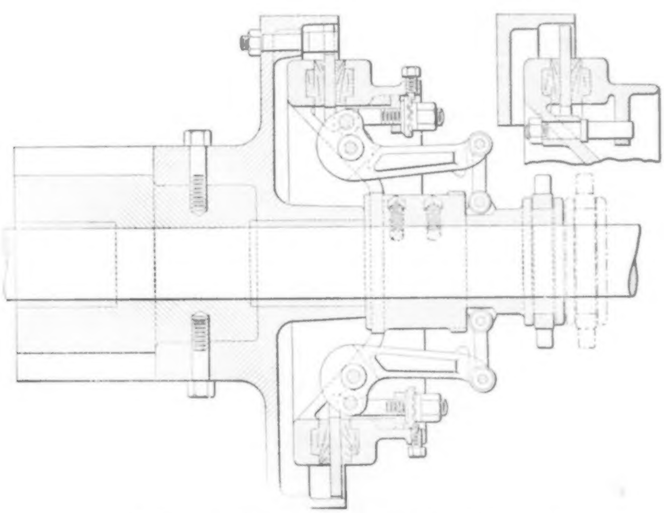


Fig. 7 Friction Clutch for Large Presses

ing the tools. A safety latch is applied to this clutch, arranged substantially like that for the Stiles clutch.

10 Fig. 6 shows a type of positive multiple jawed clutch for stamping and forging presses which run at slow speed. The clutch collar,

which is on the outer end of the shaft, is of steel, with teeth engaging the driving member, also of steel, and secured to the wheel-hub.

11 The sliding member is fitted to the shaft, which is slabbed off in the manner shown, a construction superior to keys, which become loose with the great strain brought to bear on them. The driven clutch is held out of engagement by means of the sliding bolt in the shaft, operated in the same manner as the Johnson clutch. When the foot treadle is depressed, the cam is withdrawn, and the spring on the shaft end is free to engage the clutch. The weighted lever on the right is employed to insure the separation of the clutch teeth, as the momentum of the shaft in slow running presses is not sufficient to carry the bolt on to the cam far enough to prevent the clutch teeth from clicking after the shaft is at rest.

12 In this arrangement the weighted lever is lifted by means of a cam on the shaft, and at the point of disengagement of the clutch the weight is allowed to fall with sufficient force to drive the bolt cam forward, thus separating the teeth as desired.

13 Fig. 7 shows a type of friction clutch used on large back-geared presses and quite similar to some clutches shown by the author. The friction surfaces are wood against cast iron. The cast iron disk is allowed to float on the studs in the pulley, relieving any radial strains, and the wooden friction rings are mounted on the clutch body, the clamping ring being operated by the double toggles connected to the sliding sleeve on the shaft. The shaft runs continuously with the pinion, being at rest when the press is stopped.

14 The action of this clutch, where it is applied to stop automatically at the completion of each stroke of the press, is shown in Fig. 8.

15 The cam *L* placed on the main shaft inside the gear automatically disengages the clutch. It is held in position by a bolt passing through the hub of the gear. By this means the cam can be set forward or back to insure the press stopping at the proper point.

16 The vertical rod *I* with the weight *H* attached is raised by means of the latch *J*. When the main shaft turns over, the lifter *K* drops, allowing the latch *J* to engage with the rod *I*. When *K* rises it takes the rod *I* with it, thus throwing the clutch out of action. The upper end of the latch *J* is hardened tool steel and a piece of hardened tool steel is inserted in the rod *I* for the latch to lift against.

17 In order to start the press, the treadle *P* is depressed. A link *M* is attached to the latch and connected with the bell-crank *N*, which in turn is connected with the treadle *P* by the rod *O*, the forward end of the link *M* being slotted to permit the latch to fully engage the lifter shoe in rod *I*.

18 An air dashpot Q is provided below the weight H in order to allow the weight to drop easily without shock, and an air cock is provided in the dashpot, by means of which the proper degree of cushion may be obtained.

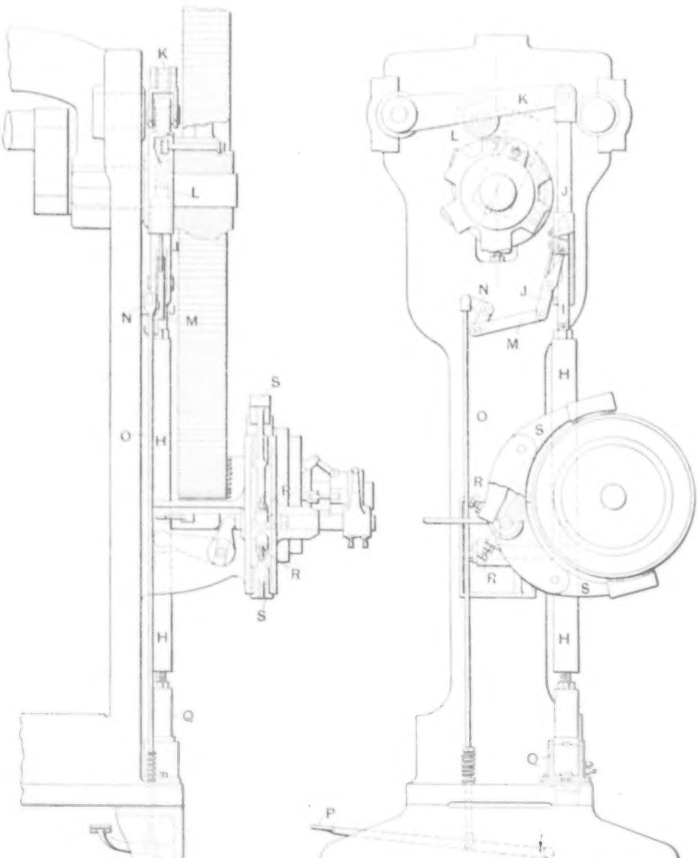


FIG. 8 APPLICATION OF CLUTCH WITH MEANS FOR STOPPING AUTOMATICALLY

19 In order to stop the press promptly and hold the shaft securely at the top of the stroke, the brake arms S are provided. The pressure on the brake arms S can be regulated by the set screws R which can be set sufficiently tight to stop the main crank in the proper position.

Mr. R. M. Gay[1] In his reference to the Sturtevant clutch, Mr. Souther mentions that as the clutch depends for its action upon centrifugal force, one could not, in the case of an automobile, speed up the engine, throw in the clutch, and jump the machine out of a mud hole. While this is true, the clutch has a peculiar action when driven by a gas engine, which makes it effective at low speeds.

2 At the instant of explosion in a gas engine the motion of the flywheel is greatly accelerated. This causes the weights of the clutch to grip harder at that instant, giving the clutch excessive torque. When you get in a mud hole and the speed of the engine drops, at each explosion the flywheel accelerates, the torque increases beyond the normal torque of the engine for the moment, and the clutch takes hold. After a fraction of a second, if the engine has not been able to pull out, the speed of the flywheel and the torque will drop, the clutch will let go and the engine will run freely, accelerating again under the next explosion. Thus the engine does not stall, but continues to pull all it can.

3 Many times I have been able to pull up grades with this clutch when the engine was running at comparatively slow speed, where with an ordinary clutch it could not be done. The effect is the same as though you threw in the clutch for a fraction of a second when the engine was exerting its extreme force and before you stopped the engine pulled out the clutch and allowed the engine to accelerate again. The centrifugal clutch does this automatically; and the engine never can be stalled.

Mr. Fred Miller[2] Clutches now in existence operate on the principle that one clutch cannot be thrown into engagement until another is thrown out, and during the interval between these two operations no power can be transmitted. We have the best illustration of this in the transmission gear of an automobile. In order to change from one speed to another, you must disengage a clutch, and thus engage a clutch which will drive at the new speed. Between these operations is a time interval during which no power is transmitted.

2 To overcome such intermittent action I have designed a clutch which continues to drive until the load is taken away from it by the engagement of a second clutch. For instance, suppose one clutch to be engaged and the train of mechanism to operate at the resulting speed. To increase that speed, simply throw in another clutch which

[1] Mr. R. M. Gay, Sturtevant Mill Co., Boston, Mass.

[2] Mr. Fred Miller, King Sewing Machine Co., Buffalo, N. Y.

will take the load away from the first one and drive the train at a higher speed without interruption to the driving action. Assuming that the train is now operating at this higher speed and it is desired to go to the lower, disengage the clutch which is driving at the higher speed and the train will fall off in speed until it reaches that of the lower speed clutch, when the power will be transmitted by the lower speed clutch.

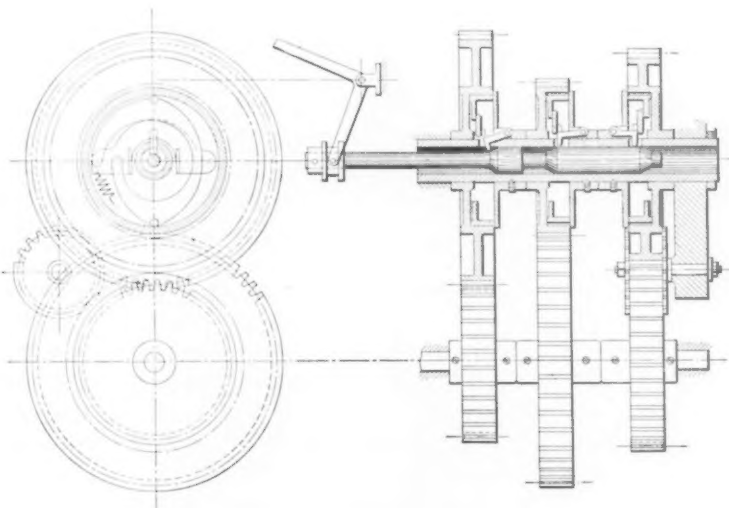


FIG. 1 MILLER CLUTCH

3 The principle which permits the higher speed clutch to take the transmission away from the lower, and *vice versa,* is that these clutches grip only when rotation is in one direction and rotation in the opposite direction immediately relieves the grip. When we engage a higher speed clutch, it is equivalent to rotating the lower speed clutch in the opposite direction, and the load is immediately taken away from that clutch and assumed by the higher speed clutch. The same principle holds true when we disengage the higher speed clutch. Rotating that clutch in the opposite direction, the speed drops until the speed at which the lower clutch grips is reached, when the lower clutch assumes the transmission.

MR. E. J. MCCLELLAN[1] The practice of The Garvin Machine Company is to use cone frictions on automatic tapping machines, with an

[1] Mr. E. J. McClellan, Chief Draftsman, Garvin Machine Co., New York.

angle of 10 deg. to the center line where the clutch is thrown in by the toggle, and 6 deg. where thrown in by a comparatively weak spring. There is little wear on the friction surfaces if the clutch is thrown in sharply. The only wear comes on the shoulder that presses the clutch in. On countershafts and friction head screw machines we use expansion rings. In this style of clutch centrifugal effect can be neutralized by turning up the clutch ring when in the expanded position so that there is an initial contracting stress in the ring. By this means we have no trouble up to some 700 r.p.m. A countershaft clutch ring of $9\frac{1}{2}$ in. diameter and $1\frac{1}{2}$ in. wide, expanded by a toggle, will resist a pull of 800 lb. on its rim.

2 One peculiarity of the Weston clutch is the accuracy with which it can be manipulated. We use a small one to control the reversing mechanism of a table on a slot milling attachment. This little clutch responds so accurately to the action of the tripping dogs, that the travel of the slide does not vary two-thousandths of an inch per stroke.

Mr. J. J. Bellman[1] The very complete paper by Mr. Souther on the subject of clutches shows that he has given the matter extended study. The work however which has come directly under his notice appears to have been in the form of clutches adapted for light high speed power transmission as found in automobile work. Clutches however are used for many forms of heavy transmission work, for which the forms most fully described are not suited.

2 The clutches best known for the heavy class of transmission, such as the winding drums of hoisting engines, or the operating mechanism of dipper dredges, are usually of the cone, band or umbrella type. With the cone clutch the practice is to force the parts together with a considerable pressure and take up the resulting end thrust by means of collars As is well known any end thrust is a serious objection, not only in frictional losses but in the means necessary to absorb it. With the cone type, the question of adjustment is a very serious one, as is frequently pointed out in Mr. Souther's paper. With heavy shafting, it is impossible to adopt the self-centering shaft, which he considers almost essential for the proper operation of the clutch. Therefore, efforts have been made to use forms of band brakes which are susceptible of easier adjustment. Much mechanical ingenuity has been applied to the means for applying the band between the two revolving elements, the most successful being by means of a steam cylinder operating on a wedge which in turn operates on links and applies the band clutch, or

[1] Mr. J. J. Bellman, 149 Broadway, New York.

a hydraulic cylinder on the driver connected directly to the band and actuated by water supplied through the center of the shaft. That these may rotate with shafts, one of the essentials for this form of applying the band, the means for applying the mechanism must be placed at the center by boring the shaft. This boring necessitates increased weight for the shaft and increased expense for the operating mechanism.

3 A method of applying the band clutch, so simple as to be of interest to those dealing with the heavier kinds of clutches in particular, has recently been devised by Mr. Gilbert H. Gilbert. The drawing shows the ordinary form of hoisting drum where the main gear is

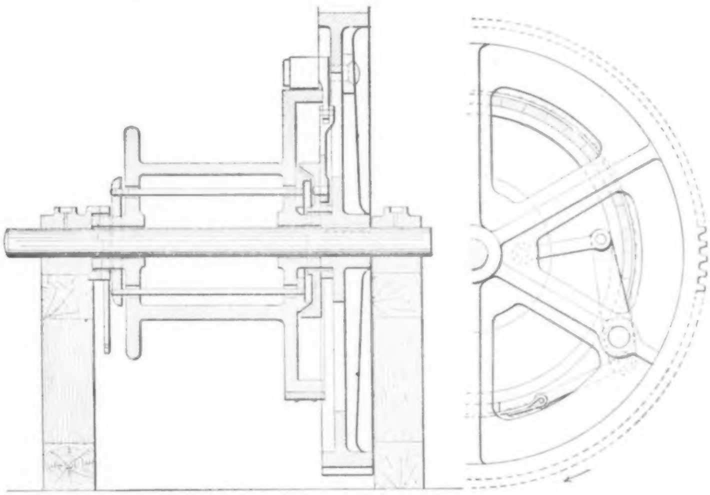


Fig. 1 Fig. 2

Means for Operating Clutches

operated continuously by a constant speed engine, and the hoisting drum is started and stopped at the will of the operator; the drum is connected to the main gear by means of an ordinary form of band brake. To tighten this brake, an auxiliary cone friction is used, one part free to revolve on the shaft, and the other attached to the hoisting drum. Pressure applied to the part attached to the hoisting drum makes it engage with the part free on the shaft, and tends to retard its revolution. A link is so connected that by the engagement of the part attached to the drum with the part free on the shaft, the band brake is applied and the drum, auxiliary clutch and gear revolve together.

Mr. B. D. Gray[1] Mr. Souther lays particular stress upon the necessity for copious lubrication and extreme care in the use of the multiple disc clutch. I am not prepared to state in a general way that lubrication is not essential to its satisfactory operation, as there may be clutches of this type, improperly designed, which require a great deal of attention in the way of lubrication. I will state, however, that the clutch with which I am most familiar, and to which I believe Mr. Souther refers in Par. 75 and 76, is not at all sensitive to the question of lubrication, and this is due to three things: first, design; second, quality of materials used; third, correct fitting.

2 I do not mean that the proper materials are very expensive or difficult to procure, but quite the reverse. The steel, for example, is commercial stock and can be obtained from any plate mill manufacturing crucible sheet. The bronze, while of a special composition, is also manufactured commercially. With proper tool equipment there is no difficulty about the fitting up in assembling, and beginning with correct design it becomes simply a commercial proposition. Oils of different viscosities for varying temperatures are not necessary, nor is it necessary to watch carefully the amount of lubrication, though care exercised in this direction naturally tends to lengthen the life of the discs. It is quite the usual thing for this type of clutch to run 25 000 to 30 000 miles with a reasonable amount of attention without renewals, and I doubt very much if even the most approved type of cone clutch will do better.

3 The extreme smoothness of operation of this particular clutch, together with its low inertia, makes it most desirable for use in conjunction with clash gears. The following data may be interesting as supplementary to Mr. Souther's short description of this type of clutch:

Inside diameter, 6.42 in. [163 mm.]
Outside diameter, 7.32 in. [186 mm.]
Area, 9.77 sq. in.
Thickness of bronze disc, 0.058 in.
Thickness of steel disc, 0.042 in.
Spring pressure 22 and 40 h.p. cars, 250.0 lb.
Spring pressure 60 h.p. cars, 300.0 lb.
Pressure per sq. in. 22 and 40 h.p. cars, 25.6 lb.
Pressure per sq. in. 60 h.p. cars, 28.6 lb.
Number of discs.
    22 h.p., 12 steel 13 bronze.
    40 h.p., 24 steel 25 bronze.
    60 h.p., 24 steel 25 bronze.

[1] B. D. Gray, Ch. Engr., Auto. Dept., American Locomotive Co., Providence, R. I.

Total area.
22 h.p., 233.5 sq. in.
40 h.p., 467.0 sq. in.
60 h.p., 467.0 sq. in.
Torque at mean radius of clutch at 1000 r.p.m.
22 h.p., 403 lb.
40 h.p., 734 lb.
60 h.p., 1101 lb.
Coefficient of friction necessary to transmit rated power at 1000 r.p.m.
22 h.p., 0.067.
40 h.p., 0.061.
60 h.p., 0.082

4 I have recently had the opportunity to try a cone clutch, which was very much the smoothest in operation of any I have seen. The clutch itself was of the ordinary type with about the usual amount of bearing surface, and with an angle of 15 deg. Its extreme softness of action apparently results from the arrangement of springs, of which there are two, one comparatively soft, and the other very much stiffer. In engagement the soft spring comes into play first, and is followed by the stiffer spring after the inertia of the car has been partially overcome. The mileage given by this clutch is remarkable, and I am informed that it has been tested under most severe conditions and without attention. It is possible that the ultimate success of the cone clutch may depend upon some such principle as this.

Mr. Frank Mossberg[1] Clutches for power presses must be so constructed as to disengage from the driving wheel and allow the driven shaft to stop in the same fixed position whenever disengagements are made. For quick running presses the clutch parts must be light enough to respond promptly when the trip lever is actuated, otherwise their inertia will so retard the action that the clutch will not properly engage with the driving wheel.

2 Perhaps the oldest clutch used for power presses is the simple jaw clutch. While this may have proved useful for comparatively slow power presses and for light work, it has been largely superseded by more improved types.

3 The principal objection to this clutch is its heavy parts, which make it slow to respond, and sometimes when the speed is up to 100 revolutions, difficulty is found in making the clutch jaws enter the driving recesses in the wheel.

[1] Descriptions of two clutches originally given in this discussion duplicated other discussion upon Mr. Souther's paper and are therefore omitted.—Editor

4 Fig. 1 shows a form of clutch used largely by several Connecticut press manufacturers. This consists of a sliding key *B* fitted to move freely in a slot or pocket in the crank shaft, and controlled by a wedge shaped cam *A* connected to a treadle or hand lever. To lock the clutch to the driving wheel the cam is released and the spring forces the key into engagement. Simplicity and low manufacturing cost are points in favor of this clutch, but it requires in practice considerable repairs.

5 Fig. 2 shows a form of clutch used on presses made by the Stiles & Fladd Press Company. The clutch pin *B* in this construction is placed in a pocket in the driving shaft in which it moves radially.

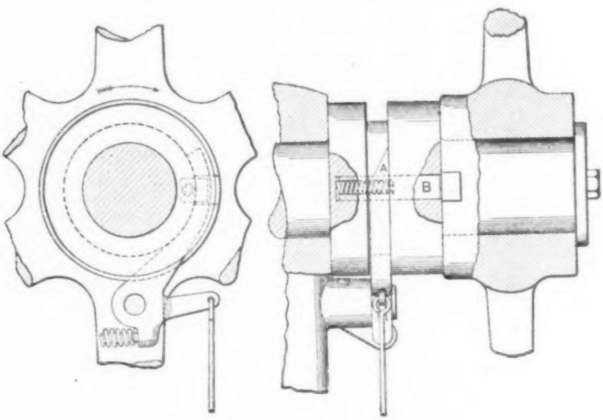


Fig. 1 Sliding Key Clutch

A spring forces this pin outward to engage with a lug in the driving wheel when the release lever is actuated. This actuating mechanism will also cause the clutch pin to recede into the pocket in which it is held by the releasing cam out of contact with the driving wheel. This clutch, having two or more contact points in the driving wheels, is a very satisfactory working mechanism and has proved very durable.

6 Fig. 3 shows a form of clutch used by the Ferracute Machine Co., and I think designed by Oberlin Smith. This clutch is used to a large extent on punching presses. It is rather complicated, especially in the form applied to larger presses. A desirable feature is that it is usually made with several contact points in the driving wheel which are of tempered tool steel. These contact points are so con-

structed that when the clutch is engaged the wheel is locked in both directions. The illustration shows the working of this clutch.

7 Fig. 4 shows a form of clutch designed by the author and intended especially for large presses. It has two large engaging pins of tempered tool steel, which act together. These are located radially opposite each other and far enough from the center for good leverage. This clutch can be made to connect at each half revolution of the driving wheel. It is one of the strongest made and has proved serviceable in practice, requiring very few repairs.

8 The clutch pins *G* connected with sliding collar *C* are mounted to move freely, and revolve with the driving wheel at all times.

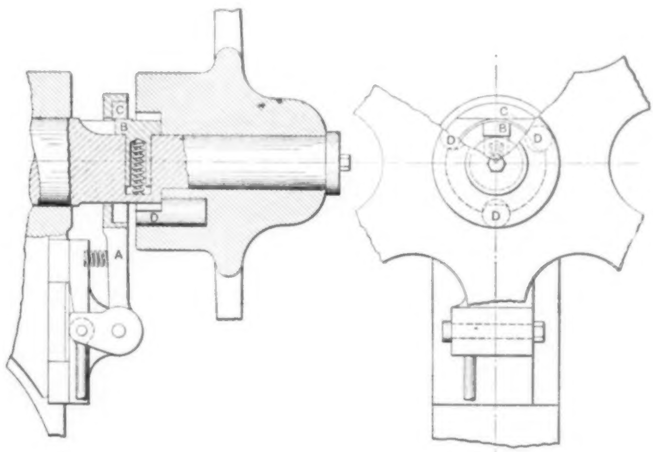


Fig. 2 Stiles and Fladd Clutch

To operate the clutch, the locking pin *E* is pulled out off the cam *D*, allowing the collar *C* and pins *G* to move into engagement with the clutch lugs *H*, which lock the driving wheel and the shaft together. When the shaft returns to its original position the clutch pins are withdrawn by the action of the cam *D*.

9 In the foregoing I have covered briefly the construction of power press clutches in general use. There are many modifications of these, but all of a class which makes a locking contact with the driving wheel and shaft at a fixed point or abutment in the wheel.

10 A total departure from all those previously described is a clutch invented by James A. Horton, of Boston, some years ago. Fig. 5

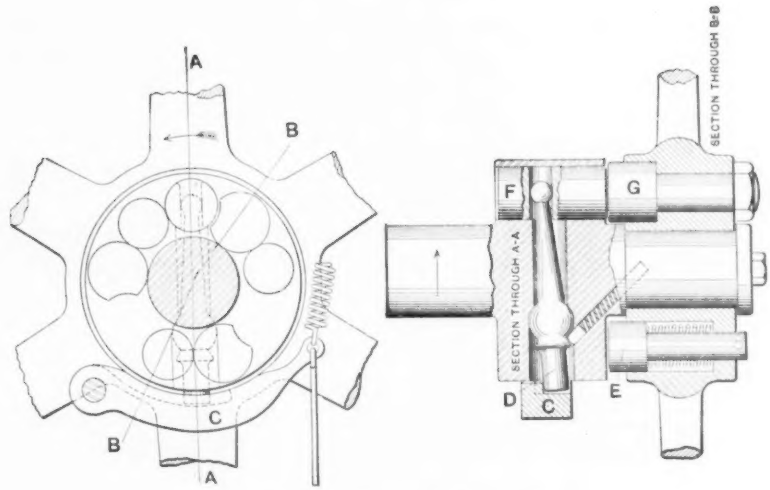


FIG. 3 FERRACUTE CLUTCH

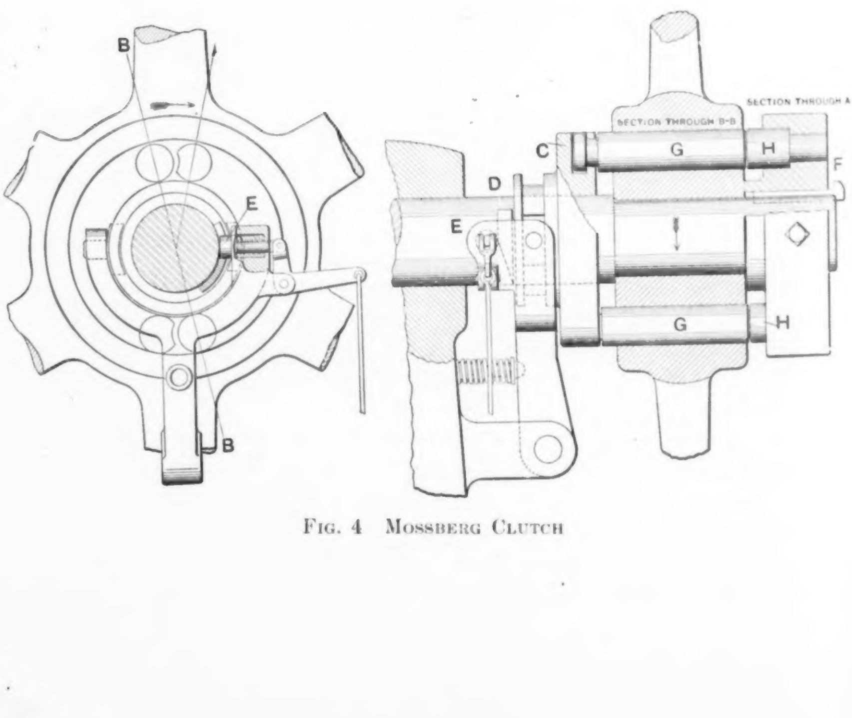


FIG. 4 MOSSBERG CLUTCH

shows this clutch, which consists mainly of a hardened steel cam *A* keyed to the crank shaft; a clutch ring *C* mounted to turn slightly on the shaft; a series of rollers *B* held loosely in slots in said ring; and a spring *G* acting on the clutch ring and causing the same to turn, carrying with it the rollers *B* toward the high point of the clutch cam *A*.

11 The balance wheel is recessed to receive this clutch mechanism, said recess being lined with a hardened tool steel ring *D*. The diameter of this recess is such that when the rollers *B* reach a point about half way between the lowest and highest point of the cam they come in contact with the clutch ring *D*, and act as wedges to lock the

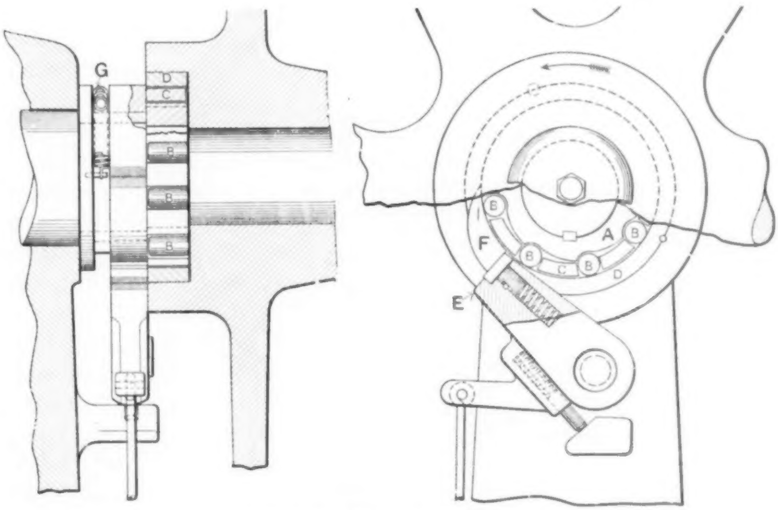


FIG. 5 HORTON CLUTCH

clutch. Release of the clutch takes place when the lug *F* fastened to the clutch ring strikes against the stop lever *E*. This will throw the clutch rollers out of engagement and allow the wheel to pass freely.

12 This I term a positive friction clutch: positive because it grips the load instantaneously when the clutch is actuated and carries it without any slip; friction, because the contact between the driving wheel and clutch rollers is simply a smooth concentric surface. As will be seen, in this clutch there are a number of locking points which act together and consequently the load is distributed. The

clutch is instantaneous in its action and can readily be disengaged at any predetermined position on the shaft.

13 A desirable feature peculiar to this clutch is that it can be released with ease under full load. In other words, with this clutch it is possible to cause the slide of the press to descend on the work, as in embossing. Release the clutch when the embossing dies act on the work, and then when the desired time has elapsed for the embossing tools to act the clutch is again engaged, and the press slide caused to return to its normal or up position.

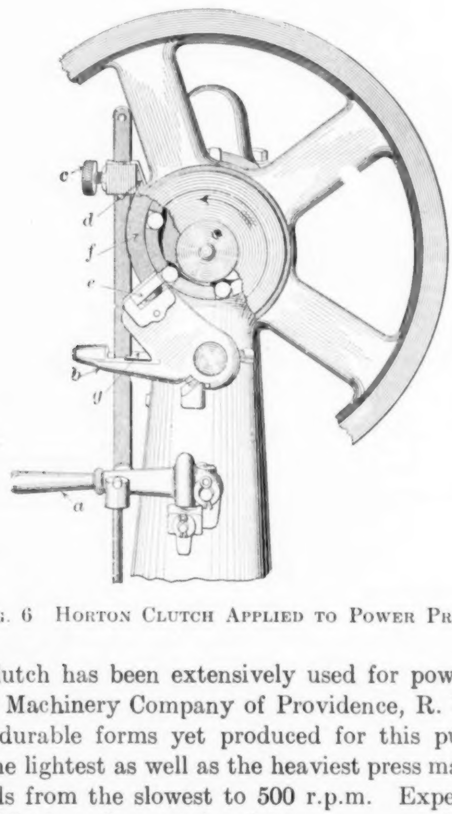


Fig. 6 Horton Clutch Applied to Power Press

14 This clutch has been extensively used for power presses by the Standard Machinery Company of Providence, R. I., and is one of the most durable forms yet produced for this purpose. It is suitable for the lightest as well as the heaviest press made and works well for speeds from the slowest to 500 r.p.m. Expense of manufacture probably has alone hindered its more general adoption.

15 The instantaneous action of this clutch when the trip lever is actuated enables the operator to run a press fitted with it faster and keep more perfect time than with any other, as is shown by the

following explanation of the working of the ordinary clutch previously described.

16 In the ordinary clutch the operator presses the treadle which actuates the fixed locking or catching key. The balance wheel is revolving around the shaft and may be just past the locking point, requiring almost an entire revolution to return to the fixed point in the wheel, while at another time the fixed point in the wheel is in such a position as to engage the locking point instantly; thus we have a stroke of the press taking practically two revolutions of the wheel in one case and only one in the next, this variation making it impossible to perform operations in unison of time.

17 By reason of its instantaneous action, and the possibility of release at any point even under load, the Horton clutch is adapted for other purposes than power presses, as for automatic drop hammers, and is the only clutch that has come under the writer's observation that could possibly be used for the requirements of this machine.

18 The clutch is located in the hub of the driving gear and is used to engage the crank shaft with this gear. To operate the drop the hand lever is tripped, causing the hammer to drop and perform the work required, and immediately upon the rebound of the hammer the clutch picks up the load without any loss of time and carries the hammer to the top position, when the clutch is disengaged and left ready for the next operation. These clutches are made in various sizes to transmit from $\frac{1}{2}$ h.p. up to 1000 h.p.

19 Fig. 6 shows the Horton clutch applied to a punching press. It also shows the tripping levers and automatic safety device which guard against the press making more than one stroke at a time, excepting by tripping the starting lever for each stroke. To make the press run continuously the clamp *d* is raised to the top of the vertical rod shown and fastened with a thumb screw *c*. The operation and function of the mechanism are so plain that detailed explanation is unnecessary.

20 The safety device described is only one of many for use in connection with a press and a press clutch and by no means confined to the Horton clutch. The high speed at which small hand-operated presses are run makes such a device very desirable for the protection of the operator's hands and fingers. Without it a press will frequently make a second stroke unexpectedly; when the operator is putting in or removing from the dies the piece which has been operated upon, and this is the time when accidents usually happen. Small presses run at a high speed, and if fed by hand, especially, should be provided

with clutches that will not start the press until the operating lever is tripped.

21 Press clutches must endure exceptional strain and abuse. Often a press with its heavy fly wheel will be brought up standing by the operator's placing the work in the die in such a way that the press cannot make the full stroke. With a fly wheel weighing seven or eight hundred pounds and 30 to 40 in. in diameter the strain on the clutch is enormous; but a clutch that will not stand this abuse occasionally without breaking is not considered desirable or practicable. Several of the clutches described are capable of meeting these conditions.

Elmer H. Neff As some friction pulleys as applied to machine tools may be of interest in this general discussion, I desire to present four styles, made and used for some years by the Brown & Sharpe Manufacturing Company. Two of these pulleys are for counter-shafts and are self-oiling. The other two are used on spindles of screw machines. I think these may be of assistance to designers in general, as showing the proper relation between the friction surfaces and the belt surfaces to insure a suitable balance between driving capacity, speed and durability. The friction surfaces of all these pulleys are metal to metal.

2 The friction pulley shown in Fig. 1 will be readily understood from the cut. These pulleys are referred to in Table 1 as old design, and are for counter-shafts of milling machines, and other machines where the speeds are moderate, and the necessity of reversal or stoppage is comparatively infrequent. There is a cast iron pulley *A* against the inner surfaces of which are brought two friction segments *BB* through the action of the levers *CC* and thimble *D*. When the thimble *D* is slid under the ends of the levers *CC* these segments are pushed apart radially until they engage the pulley. The limiting speeds which are recommended are given in Table 1, and are approximately 900 ft. per minute at the friction surfaces. Higher speeds would make it difficult for these pulleys to release on account of the action of centrifugal force on the segments *BB*. Long experience has shown that these speeds are about correct for satisfactory continuous service.

3 The friction pulley shown in Fig. 2 is the newest design for counter-shaft purposes and is used for screw machines, and in other places requiring high speed with frequent starts and stops or reversals. It consists of a belt pulley with conical friction, having the

friction surfaces at an angle of 15 deg. At the limiting speeds given in Table 2, the speed at the medium diameter is approximately 1200 ft. per minute. The operation of this pulley is, that by sliding the thimble D under the rolls in the two levers CC which are pivoted on the friction body B by pins LL, pressure is exerted at the heel against

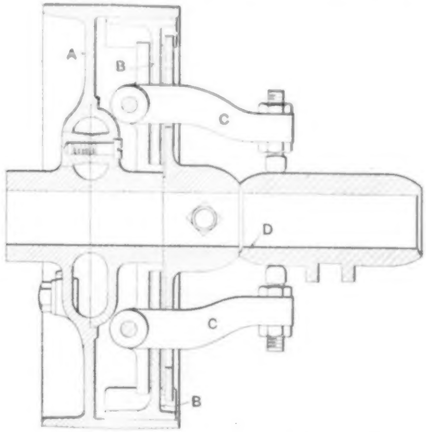


FIG. 1 FRICTION PULLEY, OLD DESIGN

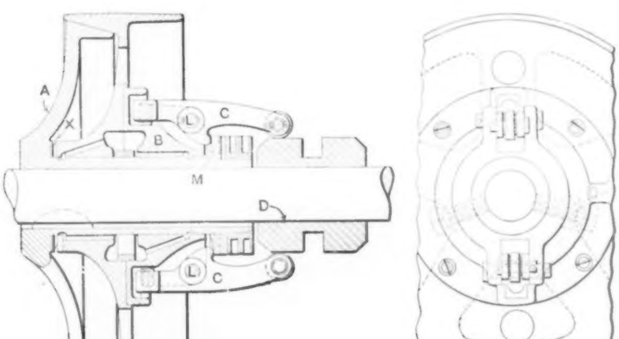


FIG. 2 FRICTION PULLEY, NEW DESIGN

the collar M, drawing the friction cone A into the pulley. A flat spring between the hub of the pulley at X and the friction A serves to separate them when the levers are released. There is a chamber for oil, and a ring oiler with passages as shown, so that the oil circulates between the bearing surfaces, and back to the chamber again.

The friction cone A is fastened to the shaft by set screws and also by a Woodruff key. Further details regarding this pulley are given in Table 2.

TABLE 1 OLD DESIGN FRICTION PULLEYS

Diameter in.	Belt in.	Highest speeds	H.P. PER 100 REV.		Diam. of friction surface in.	Width of friction surface in.	Chordal length of each shoe in.
			Single belt	Double belt			
8	2¼	450	0.59	0.93	7½	1⅞	3½
10	3	375	1.00	1.41	9⅝	1 11/16	4 7/16
12	3¼	325	1.37	1.96	11 17/32	2 3/16	5½
14	3½	275	1.58	2.30	13 15/32	2 13/32	6 5/16
16	4	250	2.36	2.92	15½	2 13/16	8
18	4½	225	2.65	3.78	17½	3 3/16	8

TABLE 2 NEW DESIGN FRICTION PULLEYS

Diameter in.	Belt in.	Highest speeds	H. P. PER 100 REV.		Largest diameter of friction in.	Width of friction surface in.
			Single belt	Double belt		
8	2½	500	0.72	1.03	7⅝	1 1/16
10	3	500	1.10	1.56	9⅝	1 3/16
12	3½	450	1.52	2.18	11⅝	1½
14	4	350	2.00	2.90	13⅝	1 11/16
16	4½	325	2.62	3.70	15⅝	1¾

4 In Fig. 3 is illustrated a fiiction clutch for use on comparatively slow speed screw machine spindles or similar work. This consists of friction surfaces on the inside of the gear or pulley to be clutched as shown in the figure at C. Inside this is the ring D with a portion cut out at point E. By the operation of a system of levers slider F is thrown in the direction desired. This slider bears on the levers GG at X, which rotates the shafts HH, on the ends of which are the cams KK. The rotation of these cams referred to, which are in the space cut out on one side of the friction rings, expands the ring into the friction pulley.

5 The system of friction pulleys for the spindles of automatic screw machines is illustrated in Fig. 4, Table 3 giving the data regarding speed and size of these pulleys for these machines. The pulleys are cast iron and the friction bodies B of bronze. The friction surfaces are at an angle of 12 deg. and metal to metal.

6 The operation of this clutch is, that by means of a system of levers the ring sleeve D, in which are two tool steel shoes H, is moved

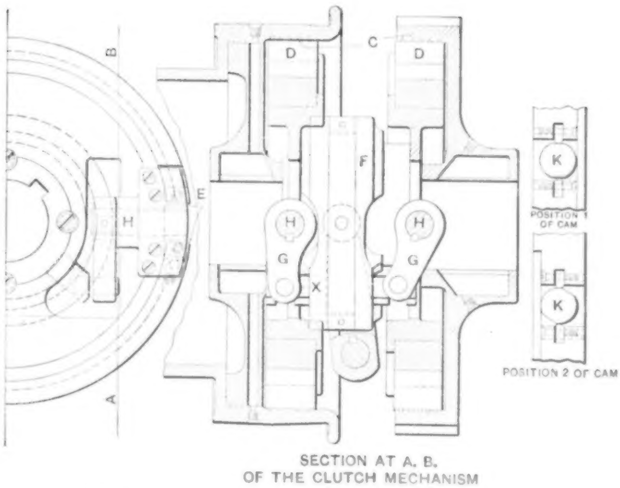


FIG. 3 SECTION OF THE CLUTCH MECHANISM

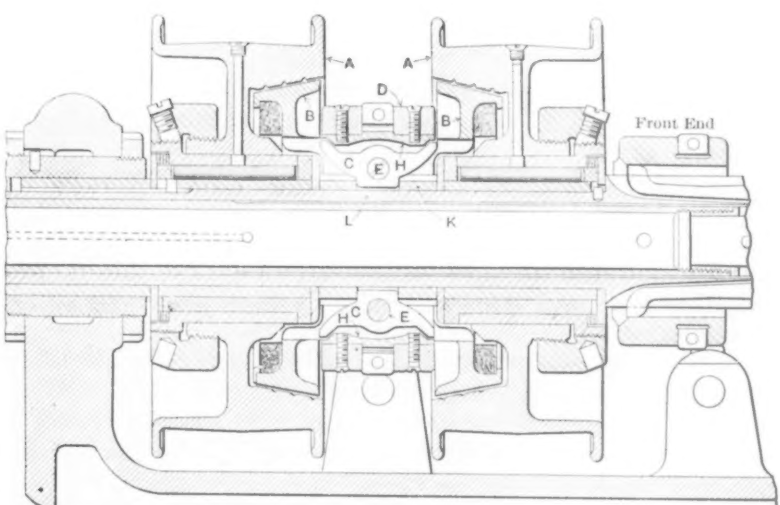


FIG. 4 SECTION THROUGH SPINDLE OF AUTOMATIC SCREW MACHINE

either to the right or left over the ends of the dogs *CC*, which are pivoted on the pins *EE*. The heels of the dogs *CC* work in slots in the sleeve *K* on the spindle *L*. When the sleeve *D* moves over the dog *C*, its action is to rotate it on the pin *E*, but as the heel of the dog *C* is against sleeve *K*, and that is immovable endwise, being rigidly attached to the spindle, the whole friction body *B* is compelled to move endwise and is thus brought into engagement with the pulley *A*. The design of the mechanism is such that, by means of cams, this friction is thrown into engagement with the pulleys instantaneously. One friction pulley is running in one direction and the other in the opposite direction. In performing a threading operation, as the die approaches the end of its cut, the lever is tripped, the sleeve *D* is thrown in the proper direction, engaging the opposite clutch, and the spindle is therefore instantly reversed.

TABLE 3 AUTOMATIC SCREW MACHINE FRICTION PULLEYS

Automatic screw machines	Largest diameter	Belt	Width of friction surfaces	Maximum speeds
No. 00	$3\frac{3}{4}$	$1\frac{1}{4}$	$\frac{1}{2}$	2400
No. 0	$4\frac{5}{8}$	2	$1\frac{1}{8}$	1800
No. 2	$5\frac{5}{8}$	$2\frac{1}{2}$	$1\frac{1}{4}$	1200

7 In the smallest size machine, the No. 00, with a maximum speed of 2400 r.p.m. we have a mean speed of about 2200 ft. per minute. As the pulleys are running in opposite directions, in order to accomplish the reverse, the friction body *B* must be thrown out of engagement with the pulley at the left, which is running at a speed of 2200 ft. per minute, into engagement with the pulley at the right, which is running in the opposite direction at a speed of 2200 ft. per minute, making the relative speed about 4400 ft. per minute at the maximum. These machines have run at these speeds for years without the necessity of replacement, showing that the friction takes hold of the pulley with practically no slip, otherwise there would be wear. It is possible to thread with these up to a shoulder, using a solid die holder with no side play whatever, and they reverse without damage to the thread being cut, the die holder, or the machine.

Mr. Charles D. Rice Fig. 1 shows a friction clutch that I have come to regard as the most reliable of any with which I have had to do. The frictional part consists of hard wood set on end in receiving

pockets, the exposed ends being forced against a smooth metallic surface of conical form. The ends of the actuating fingers are supplied with rolls, affording great releasing force with slight effort by the lessening of friction. The fingers are counterbalanced so they will not hold the clutch from disengagement at very high speeds, when applied in reverse form to what is common in automobile practice.

2 The adjusting ring or nut, against which the wedge fingers bear, affords a universal adjustment, instead of an unequalized adjustment common to some clutches in which there are three or more members to adjust. The adjusting ring is split and clamped to the threaded sleeve which holds it by means of a screw.

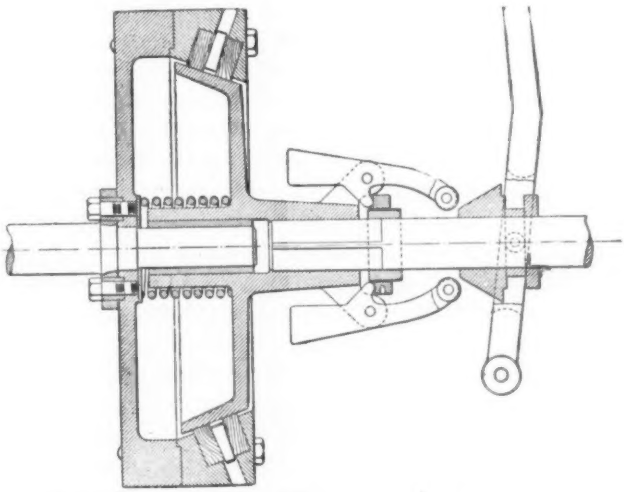


Fig. 1 Clutch with Wooden Friction Blocks

3 The wooden plugs or settings are from well-seasoned hard wood, preferably maple, boiled in oil; they are wedged in their respective positions by means of a round tapered pin which can be easily removed. Practice has proved that such a clutch will drive well when the frictional surfaces are liberally exposed to oil, enabling the clutch to start a load effectively and without undue violence while gripping.

4 Some years ago I had occasion to apply a friction clutch to a large press. The diameter of this clutch at the gripping point was 6 ft. The wood settings were substantially of the character described, and the duty of the mechanism was sufficient to require an 8-ton fly-wheel running at 100 r.p.m. The clutch was actuated several

hundred times each day and furthermore had to make an effective engagement to meet with the full resistance of the work within a rotating movement of 60 deg. The gripping surfaces were constantly exposed to a drainage of oil leading from the fly-wheel hub. It was hydraulically actuated, and operated very satisfactorily for a term of years without need for repair.

Mr. Charles Wallace Hunt A long experience in the building of several varieties of friction clutches prompts me to add to Mr. Souther's paper some data relating to the coefficient of friction of the sliding surfaces of friction clutches, and also to discuss some paragraphs in order that designers may not be misled by hasty inferences drawn from the statements therein contained.

2 A friction clutch of the type under discussion is used to connect shafts in motion, in order that the shock caused by using inelastic jaw clutches may be avoided. When the two parts of a clutch are engaged, the friction surfaces slip, and in consequence are heated in proportion to the mechanical work expended in the slipping of the friction surfaces. When shafts are connected once or twice a day, for example a line shaft in a factory, the heating effect is soon dissipated. When clutches are frequently used, as in the case of countershafts of lathes, the heating effect may be evident, but the rise in temperature is so slight that the heat is dissipated before the next engagement of the clutch. When the friction clutch is used in such work as hoisting coal, where loads of from one to several tons must be hoisted two or three times a minute for hours in succession, the dissipation of the heat is the governing factor in the design.

3 It is necessary for the designer to know in advance whether the revolving shafts are to be disengaged while the power is still being applied to the driving shaft, or at a time when no power is being applied to the driving shaft. This condition is an important factor with any type of clutch, but especially is it so with the cone type. It is the governing one in deciding upon the mechanism for operating the moving parts, and it also limits the cone angles of the friction surfaces. It is the question which must first be decided, and its neglect has been the cause of serious disappointment in many cases.

4 The centrifugal force, which in a rapidly revolving clutch tends to throw the lubricant from between the friction surfaces, is also of great importance in some designs.

5 There are but few materials now used for making the friction surfaces. These are leather, wood, cork and metal. While other

materials have been used, they vary but little in type from some one of these materials and can be safely classified with one or the other of them

6 For the design of commercial machinery, if the designer uses a coefficient of friction on iron, of 0.35 for cork, 0.3 for leather, and 0.2 for wood, he will not be disappointed in the result. These figures are derived from a long and varied experience, and are to be used for clutches where the sliding surfaces are kept slightly oiled, a condition easily maintained in ordinary service.

7 When these friction clutches are liable to be flooded with oil, the clutches will work well if there is adequate provision for disposing of the surplus oil. The following example illustrates one method. A clutch similar in type to that shown in Fig. 13 and 14, but with metal to metal friction surfaces, and a different mechanism for expanding the inner ring, has long been used. They have been in use without change of proportions for about 15 years, being thrown in and out of engagement from four to five hundred times per day. The sliding surfaces were about 14 in. in diameter, with a working face of 2½ in. With a stress on the friction surfaces of from 1000 to 2000 lb. the clutch was adjusted to slip from ½ to 1½ in. when thrown into engagement. The expanding mechanism was correctly adjusted to secure this result under the ordinary conditions of service, but when flooded with oil, as it was liable to be, the parts would slip around and make one-half to a whole revolution before the surfaces would engage. A groove was then cut across the friction surfaces longitudinal with the shaft, ⅛ in. wide and $\frac{1}{16}$ in. deep. This cross groove let the surplus lubricant out, and shortened the slip of the clutch. Additional grooves were cut across, until the slipping, when flooded, did not exceed the 1½ in. which had been fixed as the limit of movement. The distance apart of the grooves to accomplish this purpose proved to be a little over one inch, or a little less than the distance it was desired that the surfaces should slip.

8 Cone clutches have long been built in quantities for hoisting coal, pile driving and similar classes of work. Those built by leading companies in New York and Newark usually have the cone surfaces at an angle of from 20 to 22 deg. between the axis and the conical surface. These clutches require a pressure on them constantly while at work, but free themselves promptly either standing or running. The cone frictions built by another company have an angle of from 18 to 20 deg., but are always accompanied by an operating device with which to pull the friction

out as well as to press it in. The axial motion of a cone clutch of 20 deg. 36 in. in diameter, in regular service is from $\frac{1}{20}$ to $\frac{1}{16}$ in. The loads hoisted by this clutch gave about 6000 lb. pull on the friction surfaces, which were leather and cast iron. They are engaged and disengaged four or five hundred times per day. The friction leather lasts for several years.

9 Contrary to the statement in Par. 42, the engagement of a cone clutch is not a difficult problem, as large numbers of these are in daily service, operating without the slightest difficulty. The method of moving shown in Fig. 17 is admirable, and there are many other devices, the essential features of which are, absence of lost motion, and great rigidity in the operating mechanism.

10 Cone clutches used on automobiles have usually been designed with a small angle, that is from 7 to 13 deg., and with leather friction surfaces. These small angle clutches will not disengage when standing still, or when the power is not driving the machine.

11 The mechanism in current use on automobiles for withdrawing these clutches almost invariably has considerable spring, and sometimes slack motion in the joints in addition. This construction makes the cone clutch irregular in its action, and accounts for the complaints of the "fierce" action both in engaging and disengaging. If a rigid handling mechanism is used, and a larger angle cone, the difficulties will all disappear. The cone clutch then becomes admirably smooth in its engagement and disengagement.

12 The flat surface clutches used in automobiles manufactured by the De Dion Company of France, Fig. 31, and the American type made in Detroit, Fig. 17, may be considered mechanically as cone frictions of 90 deg. They work with admirable smoothness, and would also work equally well with a conical clutch if the cone angle was 20 deg. or over. The strength of the springs needed to hold the surfaces in contact increases as the cone angle increases. The cut in Fig. 17 shows a clutch that is normally off, but the same makers also build for large cars a double-faced clutch of the same type, which is normally on.

13 Leather is usually used on the inner, and iron on the outer cone. It may be interesting to note that chrome leather wears equally as well as ordinary bark-tanned leather, and is very much superior when the surfaces run hot. When the clutch is large in diameter and the amount of slipping is great enough to heat the parts, then a difficulty sometimes arises from the fact that the leather is a non-conductor of heat, and when heat is generated at the point of

sliding contact, the metal of the outer shell of the clutch becomes hotter, and consequently increases more in diameter than the inner one which supports the leather. If when a clutch is so heated it is left engaged until it cools, the outer cone will shrink more than the inner one, binding the leather so tightly that it is difficult to disengage the two parts of the clutch. For this reason, large size clutches are sometimes made with the non-conducting material attached to the exterior part of the cone, so that the inner cone will be the hotter one, then when the clutch cools, the inner cone will shrink away from the outer one and thus loosen the clutch.

14 Par. 47 refers to a form of construction that is short lived. The leather over the springs that first comes into contact will promptly wear through, and the clutch will soon be ruined by the breaking of the leather. A larger angle of cone surfaces would at once solve the problem.

15 Par. 55, 56, 57 and 58 describe clutches that are normally held in driving contact by a spring. This is the most suitable clutch for an automobile. Various operating devices may be used, but nearly all of those shown are faulty in one respect, that is, there is a springy or a loose-jointed method of moving the cone in and out of contact. To avoid irregular working of the clutch, the mechanism must be rigid and the joints accurately made.

16 The impossibility of rapidly dissipating the heat generated when heavy and rapid work is required eliminates from consideration enclosed clutches. They are frequently satisfactory on automobiles, as the service is not continuous. The oiling in this type requires care in the design. Frequently it is inadequate, or too intermittent to keep the surfaces in an equable working condition.

Mr. F. C. Billings The author of the paper holds that the cone clutch, *properly mounted*, is, all things considered, the best form of clutch for automobile purposes.

2 We have built several automobiles at The Billings and Spencer Company, and have come to realize this very point. Absolute flexibility back of the cone is essential to its good behavior, and we went to extremes in our last car to obtain this flexibility. Fig. 1 will make clear how this was secured.

3 Referring to drawing: *A* indicates the cast iron fly wheel, mounted upon crank shaft *B*; *C* the driven member of the cone clutch. The cone is leather faced with a 6 deg. angle, which is a smaller angle

than any mentioned in the paper; but it is perfectly successful and does not wedge. At D is a universal joint of the usual cross type, but in place of bearings of hardened spindles or the like, four Hess-Bright ball bearings are supplied as indicated at EE. These bearings naturally give perfect flexibility and freedom from binding in any direction.

4 We have overcome the tendency of the square telescope shaft to bind and stick at a critical moment by inserting two of the ball bearings at FF. These carry the torque with perfect freedom and give telescopic slip.

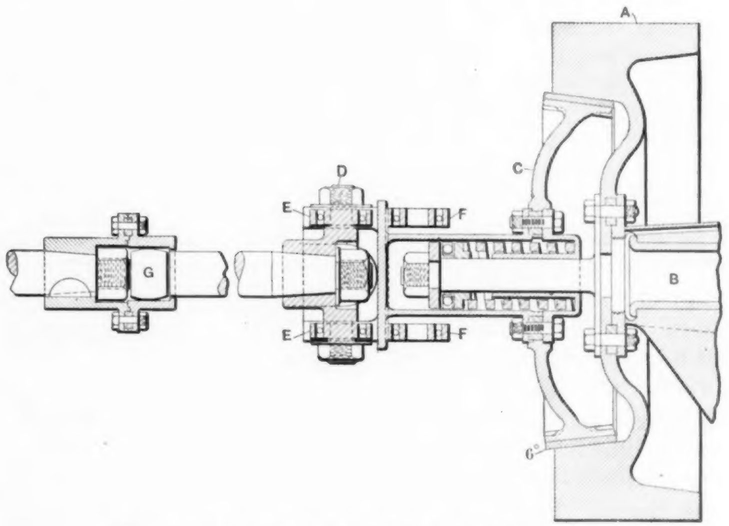


FIG. 1 CONE CLUTCH WITH FLEXIBLE MOUNTING

5 It is our belief that one universal joint is not sufficient, and we have therefore supplied at G a squared socket enclosing a nut with spherical faces as shown in the drawing. This permits full freedom of action for the universal joint ahead of it.

6 We realize that we adopted an expensive construction, but believe the expense to be worth while and that freedom from clutch trouble and absence of wear will compensate for it. A perfect acting clutch furthermore saves wear and tear on tires and the driving mechanism.

MR. FREDERICK A. WALDRON The writer was identified with the development of the multiple disc clutch, examples of which are pre-

sented by Mr. Souther, when first used on cranes in 1887–1890, and a summary of the results may be of general interest.

2 A multiple disc clutch is a good clutch to hold, but a poor clutch to release, for the following reasons:

a There is not enough lever movement to produce the clearance between the discs necessary to prevent dragging.

b As the discs are free on the shaft they are spaced more or less irregularly, causing rubbing of their surfaces. The clearance is greatest between the surfaces nearest the sliding part.

c Imperfect workmanship and heavy lubricants are a source of drag troubles, especially if the driven portion has a small static resistance when not in duty.

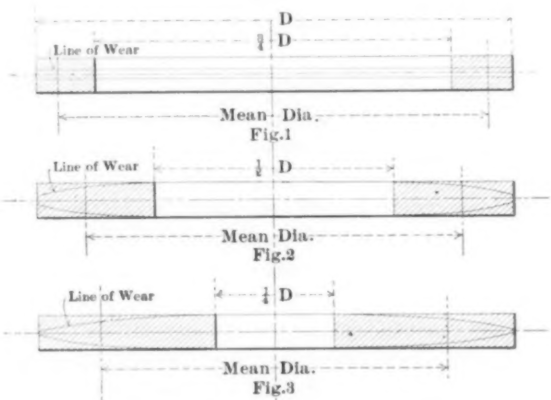


FIG. 1-3 DIAGRAMS ILLUSTRATING WEAR OF CLUTCH DISCS OF DIFFERENT MEAN DIAMETERS

3 When the driving surface is moving continuously and the driven only part of the time, and there is insufficient separation of the discs, heating and even ignition of the oil sometimes result.

4 Clutches wear most rapidly when allowed to act as cut-off couplings. The discs wear unevenly when mounted on a shaft of small diameter. Worn out discs generally appear like Fig. 3. Upsetting the keyway in discs was frequent.

5 The less the difference between the inside and outside diameters of the discs the more nearly parallel the surfaces after they become worn, and the more efficient the surfaces; assuming, of course, the area of the discs to be sufficient to meet the requirements.

6 The following rules were finally adopted in designing:

a Select materials with as great a coefficient of friction as possible. (Cast iron and elm or white wood shoes were used.)

b The mean radius of the discs should be as large as possible.

c Reduce the number of surfaces to the minimum.

d Use graphite lubrication.

e Balance the pressure due to gripping and avoid end thrust.

f Have plain lever application with two angles on sliding collar, one to grip and one to free after release.

g Insist on good workmanship, hardened points and true surfaces.

7 A clutch designed along these lines was patented by Thomas Weston, 1899 or 1890, assigned to the Yale and Towne Manufacturing Company and later sold by them to the Brown Hoisting and Conveying Company, Cleveland.

8 The following data may be helpful: Materials for friction surfaces, iron and white wood; minimum pressure, 100 lb. per square inch; maximum pressure, 200 lb. per square inch; coefficient of friction, 0.07 on each sliding surface; hand pressure on lever, 25 to 30 lb.; actual pressure on friction surfaces, 5500 to 11 000 lb.; incline on sleeve, 7 to 22; total travel of hand lever, 22 in.; ratio of hand movement to shoe movement at the time pressure is applied, 215 to 1.

9 Fig. 1 shows the wear to be practically uniform with hole $\frac{3}{4}$ *D*. Fig. 2 shows the approximate curve of worn surface with hole $\frac{1}{2}$ *D*. Fig. 3 shows wear with hole $\frac{1}{4}$ *D*. In the case of Fig. 3 the edge has worn to a thickness of only 0.002. This wear was due to the sliding when the clutch was applied gradually and the drag between the discs when not driving. Keyways and other details are purposely omitted from the sketches.

10 The foregoing results were obtained under the general direction of Mr. John W. Meserve, Mr. Warren Morse and the writer.

Mr. Hiram Percy Maxim[1] Mr. Souther points out the confusing effect of variable lubrication between the engaging surfaces. Fully as much confusion may be caused, however, by difference in surface velocity between the engaging surfaces, and especially when the surfaces are metal in both cases. Interesting examples of this may be cited from the writer's experience.

[1] Hiram Percy Maxim, Hartford, Ct.

2 In a certain automobile clutch it was necessary to transmit a force of 70 lb. at 12-in. radius. The clutch was of the expanding ring, metal-to-metal type. The internal diameter of the drum was 9.5 in. and the metal cast iron. The expanding band was a bronze casting, $\frac{3}{16}$ in. thick and 2 in. wide. It was anchored at one end and actuated at the other. The band was connected to the engine shaft and was consequently the driver. The drum was connected to the car, and consequently was the driven member.

3 This clutch, when run in ordinary light machine oil, and at a surface velocity of approximately 1000 ft. per minute, would at once pick up the stationary or driven member. If snapped in as suddenly as one could shift it, it would pick up its load very gently, and yet inside of three or four seconds time. The coefficient of friction seemed to increase inversely to the surface velocity difference. It started with the gentleness of a boat without at first materially affecting the engine speed, and then, as the speed of the driven member increased it would gently seize, bringing down the engine speed quickly, and take full advantage of the fly-wheel inertia.

4 But it so happened that 1000 ft. per min. surface speed was lower than what was practically necessary in service, in order to prevent stalling the engine when starting the automobile. The engine speed had to be about 800 r.p.m. to insure starting the car at all times, and this meant approximately 2000 ft. per min. surface speed on the part of the expanding band in the clutch. At this velocity the characteristics of this clutch were absolutely different. Where the load of 70 lb. could be picked up nicely in three or four seconds time before, it could scarcely be picked up at all at this higher speed. The coefficient of friction between the surfaces at the higher speed was but a small per cent of what it was at the lower speed.

5 In my struggles to get out of this difficulty an interesting example of the confusing effects of variable lubrication, of which Mr. Souther speaks, was presented.

6 As explained, at a surface velocity difference of 2000 ft. per min., light machine oil would prevent the metal surfaces from getting together, and this in spite of any combination of oil grooves I could arrange. The question then arose as to whether there was any other lubricant which would not tend so strongly to keep the metal surfaces separated. I thought at once of graphite as a non-liquid lubricant and one which would therefore not flow. I tried it and found to my astonishment that there was no practical velocity difference at which I could not pick up the load as quickly as I wanted. Indeed,

with excess of graphite the clutch would go to the opposite extreme and be savage. The difficulty was overcome and the solution is an interesting instance of what Mr. Souther has pointed out regarding lubrication between the clutching surfaces.

7 In leather-to-metal clutches the characteristics of engagement are entirely different from what they are in metal-to-metal. Difference in surface velocities has not as much effect. I have always considered that the reason for this was the fact that the leather being porous, any lubrication on the surfaces has an opportunity actually to sink into the fibre of the leather and escape.

8 The approximate uniformity of friction at all velocities in the case of leather is a disadvantageous feature in some automobile services. To insure against slipping, pressures have to be used which make savage starting very difficult to avoid. In racing this is especially bad, for powers are so high as to produce terrific strains. There have been many cases of rupture of gears and shafts in the transmission line of automobiles due solely to the savage engagement of the clutch. I have raced many different types of automobiles fitted with both leather faced and metal faced clutches, and the comparison of the two types is instructive.

9 In track and beach racing, the contestants are either lined up directly on the starting line and sent off at the crack of a pistol shot, or they assemble 100 yd. back of the start and line up as they bear down to the line. In either case the man with the best clutch has an immense advantage. The get away is of course the all important thing, as the one able to sweep ahead in the first few yards is able to get the pole, and on a circular track this advantage is the equivalent of several horse power.

10 With graphite in a metal-to-metal clutch I have won several races over cars able to beat me badly on a dead straightaway. In all races where I used this type of clutch my practice was to begin speeding up the engine as the starter began counting the last three or four seconds. By the time the pistol shot came the engine would be running very likely as high as 2000 r.p.m. At the crack of the pistol I would drop the clutch in with a snap—actually letting it all the way in as quickly as I could release my foot. Without the slightest jerk, it would gather headway very much as a modern high speed elevator does. It would not waste any energy in slipping the driving wheels, nor would it waste a material amount of the energy of the rapidly spinning fly-wheel. While I raced with this clutch I do not remember a case where I was not able to take the pole from my adversaries

in a very few car lengths. I personally do not believe a leather faced clutch could be made which would accomplish this.

11 The metal-to-metal clutch is slow, however, in gaining popularity. This is, in the opinion of the writer, due to its higher cost and greater manufacturing difficulties. In the simple cone type of leather faced clutch the cheapest construction is possible and in many cases a good enough action is secured. In the writer's opinion however, it can never compare with the metal-to-metal type for truly good clutch performance.

Prof. Forrest R. Jones I would suggest a modification of the first two sentences of the author's Par. 39, to read as follows: Any automobile clutch must engage *quickly*, smoothly and absolutely without shock *up to the full torque capacity of the motor with certainty of not exceeding this amount of grip by more than a few per cent at most, must not be injured by heating to a high temperature on account of the slipping that must necessarily occur when starting the car, and must be durable,* to be called a success. The italicized portion is the addition to the original sentences.

2 The torque that a given clutch will transmit depends solely on the pressure between the friction surfaces and the coefficient of friction. The latter is a variable quantity, as has been pointed out by Mr. Souther. It is generally exceedingly variable and always beyond control. The only method of regulating the torque is by variation of the pressure between the friction members.

3 To meet the requirements of a successful clutch, as just stated, *the pressure between the friction surfaces must be regulated automatically* so as to be inversely proportional to the coefficient of friction. That this automatic regulation can be accomplished successfully has been demonstrated by tests upon a clutch, a description of which the writer hopes later to present to the engineering profession.

[The auther wished to present no closure.—Editor]

No. 1189

THE SPRING MEETING AT DETROIT, MICH.

LOCAL COMMITTEE

ALEX DOW, *Chairman*

W. J. KEEP	W. S. RUSSELL
F. E. KIRBY	C. G. HERBERT
T. H. HINCHMAN, JR.	H. E. COFFIN
A. C. PESSANO	S. C. BARNES

PROGRAM

OPENING SESSION

Tuesday evening, June 23, at 9.00 o'clock

In the Convention Hall of Hotel Cadillac

Address of welcome, by Mayor Thompson of Detroit.

Response by M. L. Holman, President of the Society.

A social reunion and informal reception was held after the opening addresses.

SECOND SESSION

Wednesday morning, June 24, at 9.30 o'clock

Business Meeting. Reports of the Tellers on Election of Members and Reports of Standing and Special Committees.

SYMPOSIUM ON MACHINERY FOR CONVEYING MATERIALS

HOISTING AND CONVEYING MACHINERY, G. E. Titcomb.

CONTINUOUS CONVEYING OF MATERIALS, S. B. Peck.

THE BELT CONVEYOR, C. Kemble Baldwin.

CONVEYING MACHINERY IN A CEMENT PLANT, C. J. Tomlinson.

BELT CONVEYORS, E. J. Haddock.

Discussed by,

Spencer Miller, Melvin Pattison, Geo. B. Willcox, Charles Piez, E. H. Messiter, T. A. Bennett, Jas. M. Dodge, R. C. Carpenter, E. S. Fickes, Edw. G. Thomas, H. W. Hibbard, John McGeorge, W. T. Donnelly, H. H. Suplee, Wm. Kent, A. B. Proal.

THIRD SESSION

Wednesday afternoon, at 2.00 o'clock

CONVEYING MATERIALS, discussion continued

THERMAL PROPERTIES OF SUPERHEATED STEAM, Prof. R. C. H. Heck.

Discussed by,
H. T. Eddy, H. H. Suplee, W. D. Ennis, H. N. Davis, C. C. Thomas.

A RATIONAL METHOD OF CHECKING CONICAL PISTONS, Prof. G. H. Shepard.

Discussed by,
M. Nusim.

A JOURNAL FRICTION MEASURING MACHINE, Henry Hess.

Discussed by,
J. Royden Peirce, J. A. Brashear, C. H. Benjamin.

LECTURE

Wednesday evening, at 8.30 o'clock

CONTRIBUTIONS OF PHOTOGRAPHY TO OUR KNOWLEDGE OF STELLAR EVOLUTION, by Prof. John A. Brashear, Astronomer and Scientist. Allegheny, Pa.

EXCURSION

Thursday morning, June 25, at 10.00 o'clock

Through the courtesy of Mr. A. C. Pessano, Member of the Detroit Local Committee and President and General Manager of the Great Lakes Engineering Works, the Society was entertained by a steamer trip to the Great Lakes Engineering Works to witness the launching of the 10 500-ton steamer, the William B. Meacham, and to see the sinking into place of the fourth section of the Detroit tunnel.

FOURTH SESSION

Thursday afternoon, at 2.30 o'clock

THE SURGE TANK IN WATER POWER PLANTS, Raymond D. Johnson

Discussed by,
L. F. Harza, Irving P. Church, Morris Knowles, Chester W. Larner.

Some Pitot Tube Studies, Prof. W. B. Gregory and Prof. E. W. Schoder.

Discussed by,

Geo. A. Orrok, Sanford A. Moss, H. T. Eddy.

Comparison of Screw Thread Standards, Amasa Trowbridge.

Discussed by,

Luther D. Burlingame, F. A. Halsey.

Identification of Power House Piping by Colors, Wm. H. Bryan.

Discussed by,

G. E. Mitchell, Fred. W. Salmon, John W. Lieb, Jr., Geo. A. Mattsson.

GAS POWER SECTION

Thursday afternoon, at 2.30 o'clock

Simultaneous with the Fourth Regular Session

The By-Product Coke Oven, W. H. Blauvelt.

Discussed by,

J. R. Bibbins, C. N. Barber, C. G. Atwater, R. H. Fernald, John C. Parker, G. J. Rathbun.

Power Plant Operation on Producer Gas, G. M. S. Tait.

Discussed by,

C. W. Lummis, H. F. Smith, E. P. Coleman, C. J. Davidson, H. W. Peck, F. H. Stillman, H. W. Jones, J. R. Bibbins.

Horse Power, Friction Losses and Efficiencies of Gas and Oil Engines, Prof. Lionel S. Marks.

Discussed by,

H. H. Suplee.

A Simple Method of Cleaning Gas Conduits, W. D. Mount.

RECEPTION

Thursday evening, at 9.00 o'clock

A very enjoyable reception was held in the parlors of the Hotel Cadillac.

Friday morning, June 26, at 9.00 o'clock

Economy Tests of High-Speed Engines, F. W. Dean and A. C. Wood.

Discussed by,

Geo. H. Barrus, C. A. Dawley, Richard H. Rice, R. C. Stevens, John R. Parker, W. F. M. Goss, G. A. Young, Thos. Gray, C. H. Treat.

Air Leakage in Steam Condensers, Thos. C. McBride.

Discussed by,

Chas. A. Howard, C. L. Heisler.

Clutches, Henry Souther. Discussion continued from May meeting.

Discussed by,

Frank Mossberg, Chas. D. Rice, F. C. Billings, F. A. Waldron, H. P. Maxim, Charles Wallace Hunt, Fred Miller, E. J. McClellan, J. J. Bellman, E. H. Neff, B. D. Gray, F. R. Jones, C. R. Gabriel.

EXCURSION

Friday afternoon, at 3.00 o'clock

Excursion on the Detroit River on Steamer Britannia.

ACCOUNT OF THE MEETING

The 57th meeting of the Society was held at Detroit, June 23 to 26, with an attendance of 717. On Tuesday evening, June 23, was an informal reception, with an address of welcome by Mayor Thompson of Detroit, and a response by President Holman. Mayor Thompson expressed his pride in the professional accomplishments of the local members of the Society, and welcomed the visiting members as representatives of a profession engaged in lightening men's work and in making it possible for an ordinary man to enjoy some of the luxuries and comforts of life. In his response President Holman referred to the industrial undertakings of the city, and its many attractions, and voiced the pleasure of the visiting members and their friends at the hearty welcome extended.

BUSINESS MEETING

On Wednesday morning, June 24, was the business meeting and first professional session. President Holman called the meeting to order and read the report of the tellers of election announcing the can-

didates elected to membership in the Society. The list is published as an appendix to the report of this meeting.

The Secretary announced as follows the committee appointed by the president to nominate officers for the ensuing year: Mr. James M. Dodge, Philadelphia, *Chairman;* Mr. George H. Barrus, Boston; Mr. Fred M. Prescott, Milwaukee; Mr. Frank E. Shepard, Denver; and Mr. Newell Sanders, Chattanooga.

In making this announcement he called attention to the fact that the President in his administration had particularly recognized the national character of the organization and in choosing committees had selected members of the Society from widely different parts of the country.

The Secretary welcomed to the professional and social events of the Society the members of the other societies in convention at Detroit at that time, the Society for the Promotion of Engineering Education, the Society of Automobile Engineers and the American Gas Power Society. Announcement was also made of meetings and papers of the other societies which would interest the members of this Society.

This concluded the regular business, and under the order of new business, Professor Hutton, Honorary Secretary, made a motion providing for the revision of the report on standard methods for conducting tests of gas engines. Great credit has come to the Society from the preparation of special reports covering methods of conducting engineering research work. One of the most creditable of these is that relating to gas engine tests, but as it is now about three years old, and there has been rapid development of the internal combustion engine, the report is in need of revision. He therefore moved that the Council be requested to ask the Gas Power Section of the Society to make recommendations for the revision of the Society's code for making tests of internal combustion engines.

The motion, seconded by Dr. R. C. Carpenter, was carried.

Mr. Frank Sutton explained that one of the reasons for urging the revision of this code is the fact that authors would desire its incorporation in books to be published, and he urged that the revision be made at as early a date as possible.

The President suggested that while revising the standards for conducting tests on gas engines, it would be desirable to revise the standards for testing all engines, placing all upon the heat unit basis, which was the only proper basis for a duty test of an engine, or for engine guaranty.

SYMPOSIUM ON CONVEYING OF MATERIALS

This concluded the new business, and the balance of the session was devoted to a symposium upon the hoisting and conveying of materials. The five papers scheduled upon this subject were presented in succession and a long discussion followed. These papers were as follows:

a Hoisting and Conveying Machinery, by George E. Titcomb, Philadelphia, published in June Proceedings. This paper describes present day facilities for the mechanical transfer of materials, especially coal and ore, dealing with the intermittent class of raw material conveyors, including open air and covered coal storage apparatus, traversing and revolving bridge tramways, locomotive cranes, hoisting towers, etc.

b The Continuous Conveying of Materials, by Staunton B. Peck, published in the June Proceedings. The paper covers only the conveying of materials by continuous machines, and describes some of the types in use, giving data as to their adaptabilities, capacities, power consumed and economies effected. Following this, applications of conveying machinery are shown, with some figures relating to capacities and economy.

c The Belt Conveyor, by C. Kemble Baldwin, published in the June Proceedings. The author outlines the possibilities of the belt conveyor for handling heavy abrasive materials, and gives data for aid in the preparation of designs. He makes several claims for this type of conveyor, which his experience has shown to be well founded, holding that local conditions have an important influence on the choice of the proper width and character of belt, general arrangement, etc., and that on these matters the specialist should be consulted in order to get the best results.

d Conveying Machinery in a Portland Cement Plant, by C. J. Tomlinson, published in the June Proceedings. Conditions in cement plants are such as to test conveying apparatus to its utmost, and indicate that heavier and simpler devices should be adopted, with convenient means of recording the performances of the machinery served. The author recommends devices similar to the blast furnace, skip hoist and scale transfer car.

e The Performance of Belt Conveyors, by E. J. Haddock, published in the September number of The Journal. The results of tests upon belt conveyors are reported in this paper with conclusions and formulae bearing upon the design and operation of this type of apparatus.

THIRD SESSION

The afternoon session began with a paper by Prof. R. C. H. Heck, on Thermal Properties of Superheated Steam, published in May Proceedings, in which a graphical comparison is made of data from experiments of Knoblauch and Jakob and of Thomas upon specific heat of superheated steam under constant pressure. This shows a marked difference near the point of saturation, accounted for by the author in part by errors in interpretation by Knoblauch and Jakob. A new interpretation is made of the results of the two later investigators, bringing them into essential agreement with the work of Thomas. There is a table giving specific heat and other data relating to superheated steam.

The second paper was A Rational Method of Checking Conical Pistons for Stress, by Prof. George H. Shepard, published in the February Proceedings. In the absence of the author it was presented by Mr. George A. Orrok. The paper leads to the deduction of formulae for stress derived by methods intended to be applicable to a conical piston of any form.

The concluding paper was upon A Journal Friction Measuring Machine, by Mr. Henry Hess, published in the January Proceedings. It describes a machine in which journals can be subjected to radial loads, thrust loads, or both simultaneously, and at varied speeds. The sensitiveness of the machine is sufficient to permit an analysis of bearing friction that will show the influence of the slight sliding friction always present in bearings, besides rolling friction. Diagrams are shown indicating the relation of load and friction in radial ball bearings and in thrust ball bearings, of the speed and load in equal thrust ball bearings, and of the effect on the friction of the more usual forms of ill treatment. In the absence of the author the paper was presented by Dr. C. H. Benjamin.

LECTURE

On Wednesday evening was the delightful lecture by Dr. John A. Brashear, on Contributions of Photography to our Knowledge of Stellar Evolution. To many this was the crowning event of the several meetings. The lecture was given in the hall of the Y. M. C. A. before a large audience, which the speaker had at his command until his last word was spoken. Many photographs were thrown on the screen, beginning with views of astronomical apparatus, and following with views

of constellations, the moon and the planets; and of greater beauty than all, photographs of nebulæ, the existence of which had never been suspected until detected by the photographic plate.

FOURTH SESSION

The session on Thursday afternoon, June 25, opened with a paper on The Surge Tank in Water Power Plants, by Mr. Raymond D. Johnson, published in the June Proceedings. The paper deals with the momentum of flowing water in long pressure pipes for the supply of hydraulic turbines for impulse wheels, and the control of rate of flow for the speed regulation of water wheels without harmful results or waste of water through regulating valves or deflecting nozzles or by-passing it. This is accomplished by a surge tank near the down stream end of the pressure pipe, which may be either under atmospheric pressure simply, or under compressed air. The paper treats the subject mathematically, and gives formulae for proportioning the sizes of surge tanks under various conditions, besides treating of a novel device called by the author a differential regulator, by which the diameter of the tank may be reduced, thereby lessening its cost.

Some Pitot Tube Studies, by Prof. W. B. Gregory and Prof. E. W. Schoder, published in the May Proceedings, considers the distribution of velocities and pressures of flowing water as determined by experiment in straight and curved portions of a pipe, by the aid of the Pitot tube. A study is made of the indications of a Pitot tube, both with the impact opening facing the current and the reverse, and there is given a convenient method for making Pitot tube traverses in straight pipe.

A brief paper, A Comparison of Screw Thread Standards, was contributed by Mr. Amasa Trowbridge, and published in the June Proceedings. It contains a diagram showing to what extent present recognized standards for screw threads harmonize or may be combined. The paper was read by Mr. L. D. Burlingame, who also contributed a discussion proposing formulae for a standard fine pitch screw system, forming a connecting link between machine screw sizes and United States standard threads used on the larger sizes of bolts. The paper was further discussed by Mr. F. A. Halsey, who contended that a standard for fine pitch screws was desirable and moved that a committee be appointed to investigate the subject. The motion was carried.

The next paper in order was on The Identification of Power House Piping by Colors, by Mr. William H. Bryan, published in the June Pro-

ceedings. It advocates the adoption of a system for coloring pipe lines, with further subdivision, if required, by giving the flanges a different tint, by means of which the different pipe lines in modern power plants may be distinguished and confusion avoided. The author presents schemes for coloring pipe lines of any size and believes the time is ripe for the establishment of a uniform standard.

At the close of his discussion Mr. Lieb offered a motion, seconded by Mr. L. D. Burlingame, that the suggestions made in the paper as to the formulation of a code of symbols for power house piping be referred to the Council of the Society for consideration. The motion was carried.

GAS POWER SESSION

A session of the Gas Power Section of the Society was held simultaneously with a regular session on Thursday afternoon. Dr. Charles E. Lucke, president of the Section, presided.

The first business was the reports of the standing committees. The secretary of the Section, Mr. Suplee, presented a progress report for Mr. Henry L. Doherty, chairman of the Section's Meetings Committee, stating that papers were in hand for a meeting in New York in October.

The secretary also presented a report for Mr. Robert T. Lozier, chairman of the Section's Membership Committee, calling attention to the fact that registration is necessary for members of the Society to be enrolled as members of the Section, and also urging all others interested in the subject of gas power who have not joined the Section to enroll as affiliates.

The Section's Committee on Standardization made a progress report, including extended communications from Mr. J. R. Bibbins, Mr. H. F. Smith and Mr. Arthur West, members of the committee, together with a communication from Mr. J. B. Klumpp, a member of the Gas Engine Committee of the National Electric Light Association.

The secretary reported the action at the business meeting of the Society to the effect that the revision of the Society code for testing gas engines had been referred to the Council, with suggestions that its revision be placed in the hands of the Gas Power Section.

GAS POWER PAPERS

The Section then proceeded to the discussion of professional papers as follows:

The By-Product Coke Oven, by Mr. W. H. Blauvelt, published in March Proceedings. This paper discusses the advantageous production of coke in retort ovens for the purpose of securing the valuable chemical by-products. Its principal interest to the Section lies in the discussion of the production of power gas in connection with the coking operation, and the paper contains tabulated data of the yields and character of the gas produced from various coals and the application for use in gas engines.

The next paper was upon Power Plant Operation on Producer Gas, by Mr. G. M. S. Tait, published in the June Proceedings.

Mr. Tait's paper refers especially to the desirability of using a producer gas containing little or no hydrogen, the combustible consisting almost entirely of carbon monoxid. The producer described by Mr Tait dispenses entirely with steam or moisture in the blast and utilizes a portion of the exhaust gases of the engine supply below the grate to dilute the air and regulate the combustion.

The next paper was upon Horse Power, Friction Losses and Efficiencies of Gas and Oil Engines, by Prof. Lionel S. Marks.

This paper is principally a discussion of the question as to whether the power required to operate compression pumps and other auxiliary parts of the engine should be deducted in computing the indicated horse power. Reference was made to the extent of the discussion on this subject three years ago in the Society of German Engineers, and it was agreed that the whole matter should properly come before the Standardization Committee for consideration. The secretary was requested to prepare an abstract of the German discussion for the use of the members and of the Standardization Committee.

The last paper was upon A Simple Method of Cleaning Gas Conduits, by W. D. Mount.

RECEPTION

On Thursday evening a reception, with dancing and refreshments, was held at the Hotel Cadillac.

CONCLUDING SESSION

The last session was on Friday morning, June 26, and the first paper was upon Economy Tests of High Speed Engines, by Mr. F. W. Dean and Mr. A. C. Wood, published in June Proceedings. It reviews tests upon eight high speed engines, which had been in use a considerable

length of time. The object of the tests was to determine under the conditions of actual running operation the steam per indicated horse power per hour and per kilowatt hour, and also the efficiency of the generating sets as shown by the ratio of the indicated horse power to the electrical horse power at the switch board. The engines were not put in order for the tests.

The second paper, upon Air Leakage in Steam Condensers, by Mr. Thomas C. McBride, published in June Proceedings, advocates measuring by air pump displacement and temperature the amount of air passing through a condenser, and comparing this amount in different condensers on the basis of its ratio by volume to the steam being condensed. The paper recommends that in testing condensers engineers measure the amount of air on the basis suggested, in order that the amount of air anticipated by ordinary practice may be established. Tests are cited showing widely different air leakages under different conditions.

The concluding paper on Clutches, by Mr. Henry Souther, had been presented at the New York monthly meeting in May. The discussion was continued at this time, and at the invitation of the Society members of the Society of Automobile Engineers participated in it.

This concluded the professional papers, and resolutions were then offered by Mr. C. W. Hunt, and seconded by Mr. Charles Whiting Baker, extending the thanks of the Society to the members of the local committee and others who had provided so enjoyable a time for the visiting members. The resolutions were as follows:

WHEREAS, The American Society of Mechanical Engineers in convention assembled at Detroit, June 26, 1908, desires to express its appreciation of the many hospitalities extended to its visiting members and friends by its hosts, the local members and their friends of Detroit; and to all who, by untiring efforts, have made the Spring Meeting of 1908 so extraordinarily pleasant, and so profitable an occasion.

Be it Resolved, That the Secretary be instructed to extend the thanks of the Society and to express the deep appreciation of its members and guests to the chairman and members of the local committee and especially to the ladies of the local committee for the splendid entertainment provided; to the Great Lakes Engineering Works and its president, Mr. A. C. Pessano, for the arrangements connected with the launching; to Mr. F. E. Kirby and the engineering staff of the Detroit Tunnel; to the Detroit, Belle Isle and Windsor Ferry Company for the use of an excursion steamer; to the several manufacturing plants of the city, so freely opened for inspection; and to Prof. J. A. Brashear for his lecture, which afforded so delightful an evening's entertainment.

These resolutions were passed by unanimous vote, and the meeting was then declared adjourned.

Entertainment

The entertainment provided by the local committee and others interested in the Society will long be remembered by the visiting members and their guests because of the solid enjoyment and continuous round of pleasures which it afforded.

THE LAUNCHING

The launching of the Daniel B. Meacham, which had been anticipated so much, was a beautiful sight. The members were carried by steamer to Ecorse and when the boat drew up alongside the docks of the Great Lakes Engineering Works everything was in readiness for the launching. The army of men stationed beneath the hull of the Meacham to drive in the wedges and raise the boat so she would be supported on the shoes which were to carry her down the ways, promptly began their work. The blows in rapid succession sounded like a battery of machine guns. When at last the eight cables holding in place the eight timber triggers which retained the boat at the top of the ways were severed simultaneously by eight axemen, the hull slid gracefully into the water.

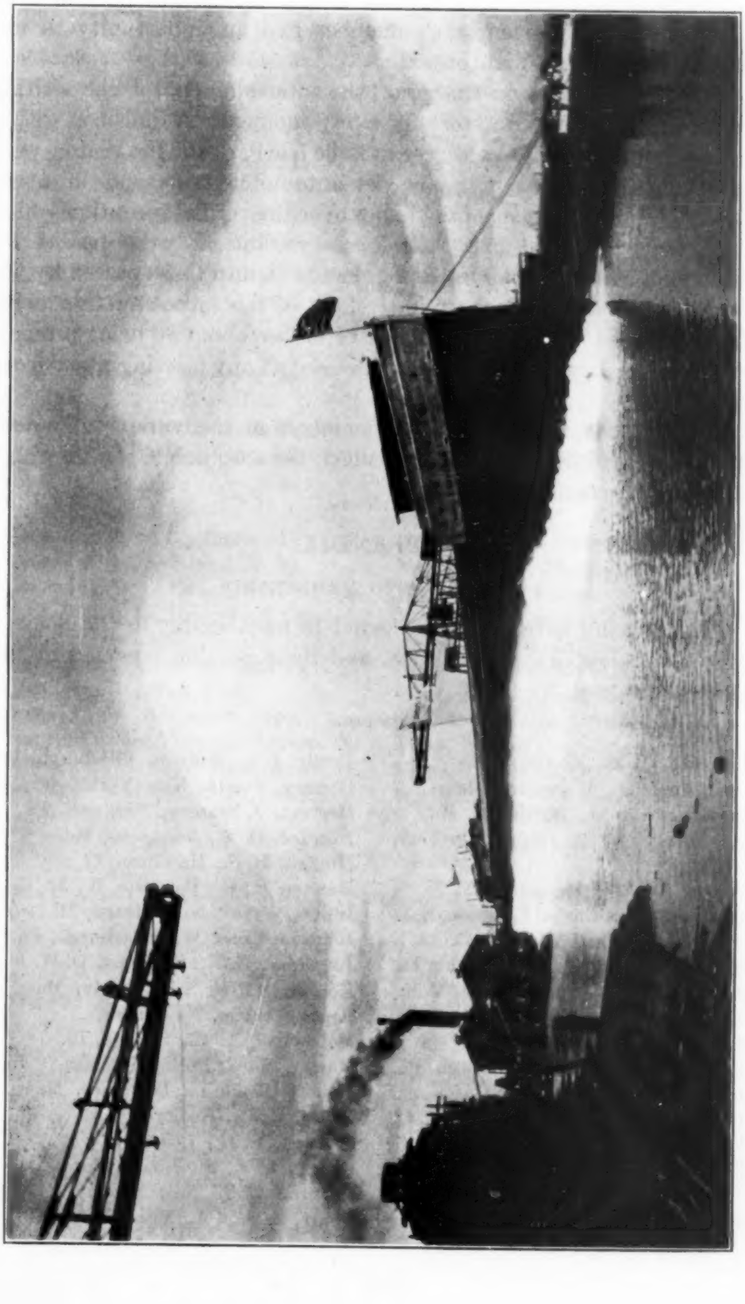
The launching was arranged to occur at the time of this meeting through the untiring efforts of Mr. Antonio C. Pessano, member of the Detroit local committee, and president and general manager of the Great Lakes Engineering Works. The date of the launching had been advanced two weeks in order to have it occur at the time of the meeting, which had necessitated night and day work to make everything in readiness.

It was the 49th launching at the Great Lakes Engineering Works and Mr. Pessano pronounced it the best one of them all.

On the return from the launching a view was had of a section of the Detroit tunnel as it lay near the shore, preparatory to sinking it to the bed of the river.

OTHER SOCIAL FEATURES

The excursion on Friday afternoon on the steamer Britannia, tendered by the Detroit, Belle Isle and Windsor Ferry Company, through Mr. F. E. Kirby, member of the Detroit local committee, was the important social event, and consisted of a sail to and from Bois Blanc Island, with a fine dinner on the island at the Casino. The real bene-



LAUNCHING OF THE DANIEL B. MEACHAM

fits to be derived from a convention were most completely secured on this delightful trip, where each member had an opportunity to exchange experiences with others.

The Detroit ladies in charge of the entertainment of the visiting ladies had something in store for every moment. A luncheon with a most tempting menu was served at Belle Isle Park and the visitors were entertained at the country club, by automobile trips and in other ways. The enthusiasm of the ladies over the cordial reception which they received was so great that special resolutions were passed by them expressing their appreciation, in addition to those passed by the Society itself at the final session. Part of the success of this meeting was due to the large attendance of ladies, about 80 being present, which is the largest number ever present at any meeting away from New York.

Much interest was shown by the members in the various manufacturing plants of the city, and particularly the automobile plants, which many took occasion to visit.

APPENDIX

ELECTIONS TO MEMBERSHIP

The following were declared elected to membership in the Society upon the ballot of June 25, 1908, and their election reported at the Detroit meeting.

MEMBERS

Andersen, J. M., Boston, Mass.
Back, John R., Worcester, Mass.
Basshor, Chas. H., Baltimore, Md.
Beckstrand, E. H., Salt Lake City, Utah.
Benton, Geo. H., Metuchen, N. J.
Brown, Herman Elisha, Kingston, N.Y.
Browning, V. R., Cleveland, O.
Burton, Isaac Francis, Philadelphia Pa.
Carpenter, Harold Eugene, New York.
Chalkley, H. G., Lake Charles, La.
Clark, Herbert H., Los Angeles, Cal.
Cox, Frederick W., E. Pittsburgh, Pa.
Dodge, A. C., Chicago, Ill.
Dunlap, Thaddeus Cox, Columbus, O.
Ferrier, Walter, Donora, Pa.
Gildersleeve, David H., West New Brighton, N. Y.
Gill, Lester Willis, Kingston, Ont.
Greth, J. C. William, Pittsburgh, Pa.
Henney, David, New York.
Herbert, J. Stanley, Nazareth, Pa.
Howard, O. Z., Annapolis, Md.
Hughes, B. S., Hamilton, O.
Hughes, R. G., Paterson, N. J.
Jewett, Frank N., Chicago, Ill.
Johnson, Chas. W., Pittsburgh, Pa.
Junghans, E. K., St. Croix, D. W. I.
Kellogg, Alfred S., Waverly, Mass.
Kroto, George, New York.
McDougall, A. H., Harvey, Ill.
Martin, Mack, Pullman, Wash.
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SUMMARY

Election to full membership	51
Election to Associate Grade	22
Election to Junior Grade	48
Promotion to Members Grade	16
Promotion to Associate Grade	2
Total number declared elected	139

No. 1192

HOISTING AND CONVEYING MACHINERY

INTERMITTENT TYPE OF APPARATUS

By George E. Titcomb, Philadelphia

Non-Member

The development of modern hoisting and conveying machinery is an interesting subject from an economic, as well as an engineering standpoint. In 1907 there were taken from the ranges adjacent to Lake Superior and transported to furnaces in the Pittsburg and Ohio district 42 000 000 tons of iron ore. Following the same route but in the opposite direction, there were mined in Pennsylvania, Ohio, and elsewhere 17 000 000 tons of anthracite and bituminous coal, and shipped to the Northwest, the vessels loading usually at the same ports at which they discharged their cargoes. The labor cost of transferring such great quantities of these materials between vessels and cars has become an important consideration, and in some respects the time spent in loading and unloading the vessels costs more.

CARGO STEAMERS AND LOADING WHARVES ON GREAT LAKES

2 Efforts to secure economical handling have developed machinery combining mechanical simplicity with great capacity and economy, and have made possible the steamers of special design now engaged in this business, and the lowest freight rates known in the world. Many of the modern vessels engaged in this trade have a length of 570 ft., a beam of 58 ft., and a draught of 18 ft., and are provided with hatches spaced 12 ft. centers. When these hatches are removed the cargo is practically uncovered and can be unloaded, with little or no hand labor being necessary for trimming. The ore coming in cars from the mines is shipped from about thirty-two large wharves at ports on Lake Superior and Lake Michigan. Some of these piers are 2000 ft. long, with tracks arranged so that four large steamers can take

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

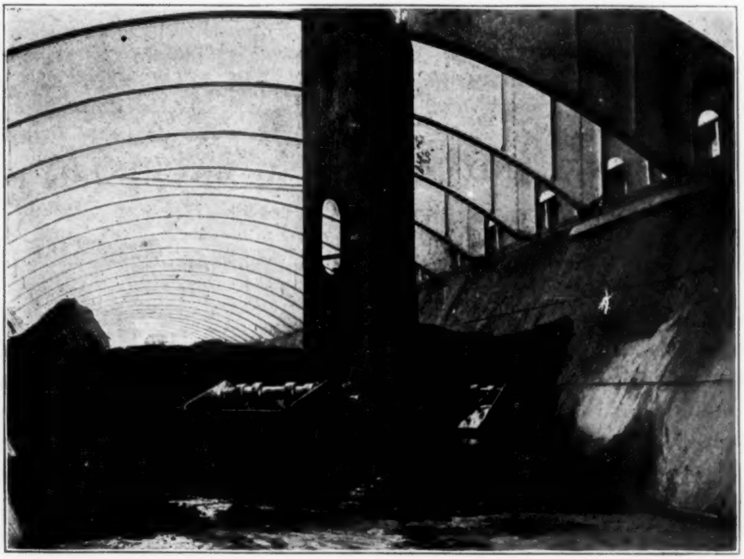

Fig. 1 Bucket of Hulett Automatic Ore Unloader in Hold of a Vessel

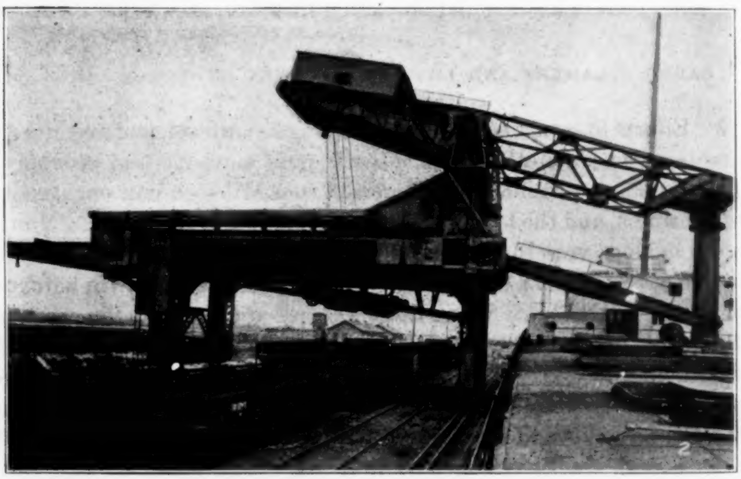

Fig. 2 Hulett Automatic Ore Unloader

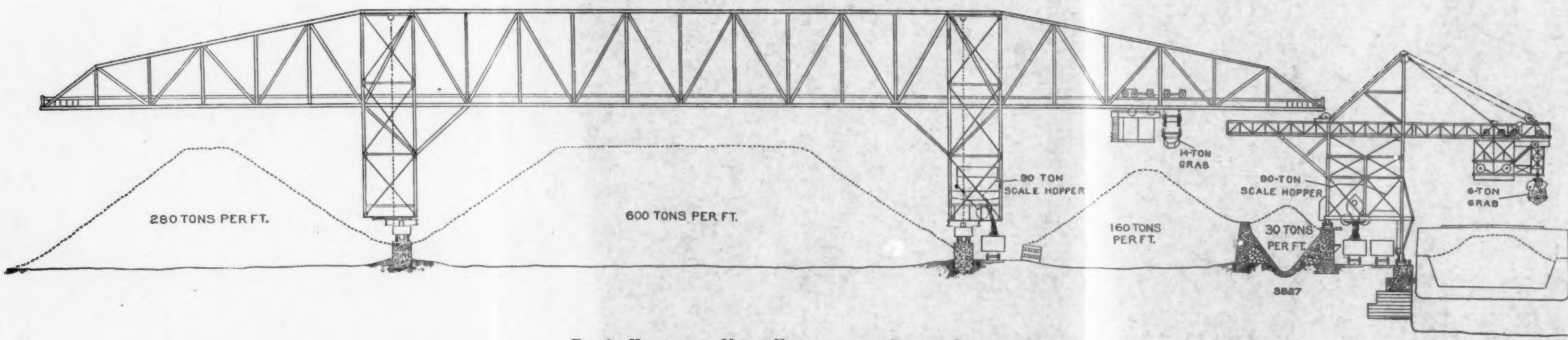

FIG. 3 HOOVER AND MASON UNLOADING AND STORAGE SYSTEM

on cargo at once. The ore is run directly into the holds from the cars or storage pockets, and carried by gravity through chutes which are lowered into the hatches.

ORE UNLOADING MACHINERY ON GREAT LAKES

3 In unloading ore from lake vessels, on account of the difficulty and cost of shoveling by hand, three modern types of machinery for performing this work, practically without manual labor, have been installed. The first is the Hulett automatic ore unloader shown by Fig. 1. This machine operates a self-filling bucket of large capacity mounted at the end of a rotating vertical leg; the bucket having sufficient spread to reach more than half way from the center of one hatch to the center of another. It also travels lengthwise of the hatch to the sides of the boat, and, consequently, commands a large percentage of the cargo. The bucket is closed and removed from the hold under the control of an operator located in the lower end of the leg where he has an unobstructed view of the cargo and vessel. The bucket discharges its contents into a bucket-car for delivery to railroad cars located under the gantry of the machine for shipment to blast furnaces, or it may be drawn out to the end of the cantilever and dumped so as to form a pile to permit re-handling by buckets to storage pile.

4 Other types of machines used for this purpose are shown by Fig. 3, a Hoover & Mason unloading and storage plant, and Fig. 4, a Brown Hoisting Machinery Co.'s unloader and storage bridge. On these machines, the bucket, instead of being fixed at the end of a vertical leg, is rope-suspended from a hinged cantilever over the vessel, and is operated by a man-trolley provided with means for rotating it when in the hold, so as to more completely command the cargo, or to remove the bucket from the boat and deliver its contents either into cars under the gantry, or onto stock pile. From the latter it is handled by a similar bucket, but of larger capacity, mounted on a movable bridge tramway provided with double cantilevers, as shown by Fig. 5.

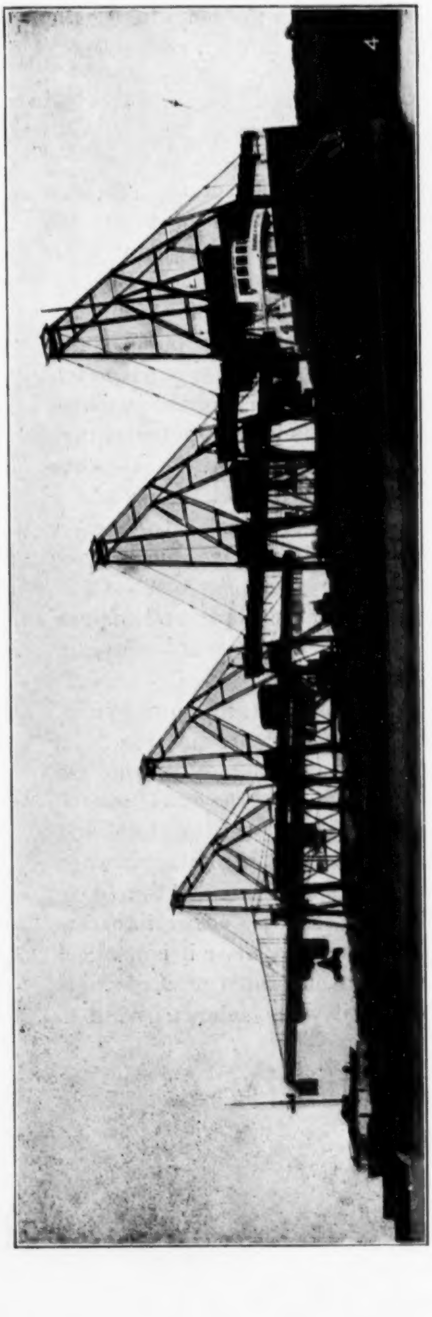

FIG. 4 BROWN HOISTING MACHINERY CO.'S UNLOADER

FIG. 5 ELECTRIC MAN-TROLLEY BRIDGE TRAMWAYS

FIG. 6 ORE UNLOADER AT BLAST FURNACES

ORE STOCKING MACHINERY AT FURNACES

5 The iron ore received in cars at the blast furnaces is discharged by means of car dumping machines, or through the bottom of the cars into track hoppers, or taken directly from the cars by the self-filling bucket with which the machines are equipped, as shown by Fig. 6. This machine is electrically operated and its bucket, which holds 5 tons, has a hoisting speed of 300 ft. per min. and a trolley speed of 800 per min., and an average capacity, for one hour, of sixty trips from track hopper to stock pile, or from stock pile back to cars. For moving the bridge back and forth over the stock yard, power-driven mechanism is provided to operate it along its tracks at the rate of 100 ft. per min.

ANTHRACITE COAL STORAGE MACHINERY

6 Anthracite coal, owing to the fluctuations in demand and production, is stored in large quantities between mine and market, making possible continuous, instead of spasmodic working of the mines, and also providing convenient bases of supply. A prominent installation is shown by Fig. 7, a general view of a modern plant of the "Dodge System." The immense quantity, over 480 000 tons, is divided into eight separate piles, each pile having a capacity in excess of 60 000 tons. The standard machine for storing the coal, which is received from the mines in cars, consists essentially of a chain and flight conveyor supported by a shear truss, constituting a trimming machine which piles the coal without breakage, delivering it upon the ground or at the ascending apex of a conical pile as it is formed.

7 The reloading of the coal is begun by a pivoted open-side conveyor which runs on circular ground-level tracks extending between and over the area to be covered by the piles. This conveyor works against the edge of the pile, as shown by Fig. 8, and follows up this receding edge as the coal is removed; it is operated by power and is so devised as to be fully under the control of one man. All of the coal is tributary to the reloading conveyors and is carried without intermediate transfer up the incline to a reloading tower (as shown in the background of Fig. 7), where it can be delivered either directly into cars, or screened and then delivered into cars.

8 In detail the "Dodge System" as usually applied to open air storage on leveled ground consists of two trimming machines and the pivoted ground conveyor which travels radially between them, form-

Fig. 7 Anthracite Coal Storage Plant—"Dodge System"

ing one unit or group. A coal storage plant consists of a number of these groups, which may be of equal or varied capacities. The shear trusses are fixed to correspond with the angle of repose of the coal,—the front truss of each group, *i. e.*, the truss adjacent to the track-hopper or receiving-point, containing the conveyor. This conveyor runs in a trough having for a bottom a continuous steel ribbon about 12 in. wide, which is wound upon a drum at the foot

Fig. 8 Pivoted Open Side Conveyor for Reloading Coal

of the truss, and is paid out only as demanded by the formation of the pile; this feature provides a gentle discharge of the coal and prevents breakage.

9 Modifications of the standard "Dodge System" have been made to meet local conditions; hill-side storage has been increased by mounting the trimmer-conveyors on cantilever trusses, so as to command fully the storage area after the initial and comparatively small gravity capacity has been reached. Where a continuous pile of

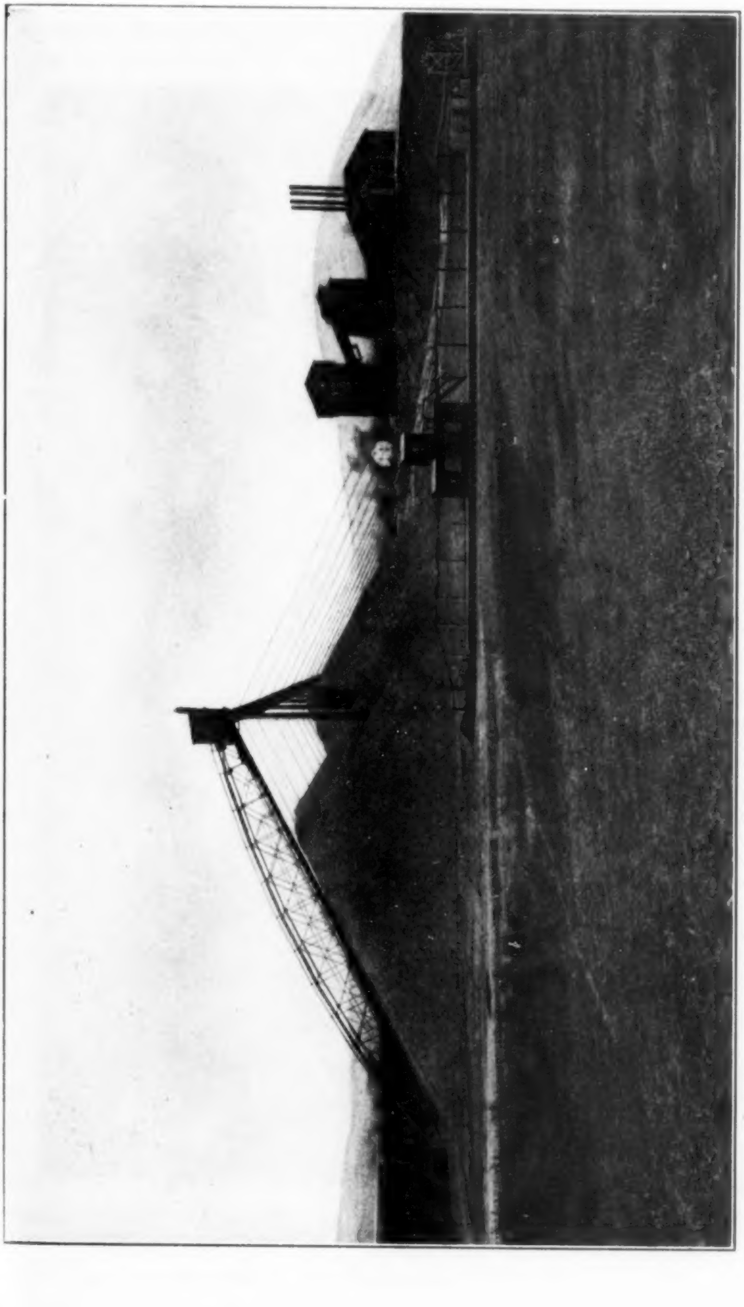

FIG. 9 PLANT WITH TRIMMER AND RELOADING MACHINE OF TRAVERSING TYPE

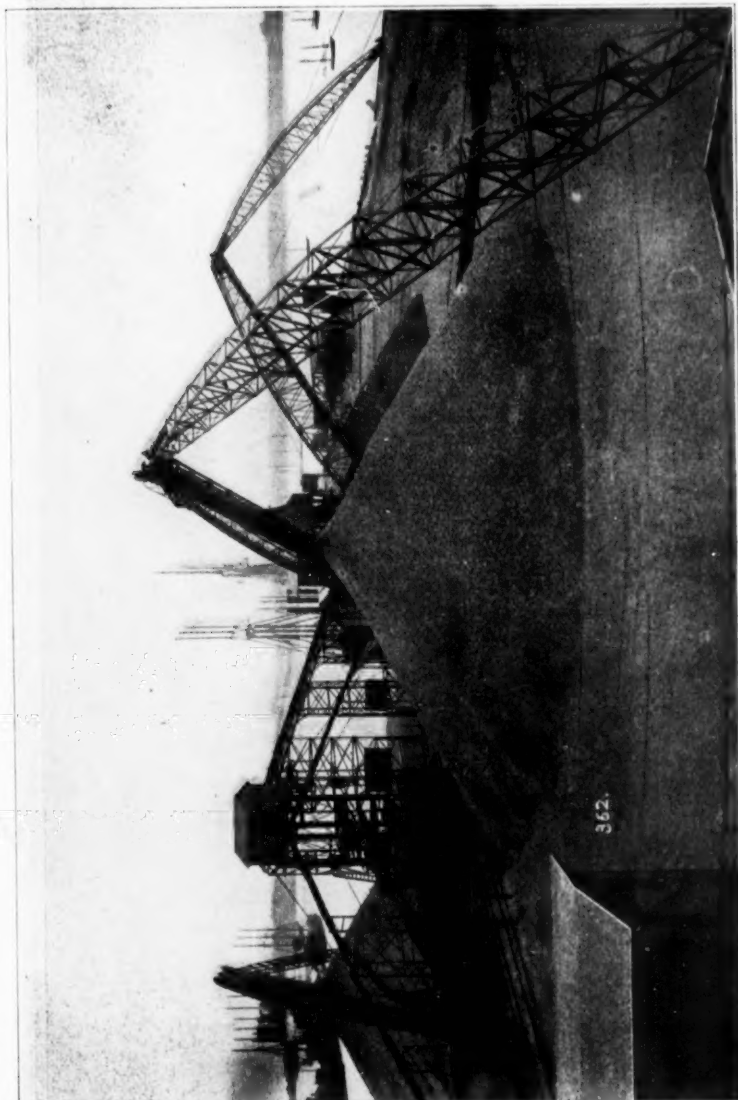

FIG. 10 ANTHRACITE HANDLING AND STORAGE MACHINERY SITUATED ON WATER FRONT—"DODGE SYSTEM"

maximum capacity is needed, a traveling trimmer is used, supported at the foot on a power-driven truck, and at the head on a column-supported track. Stocking out, or storing, is accomplished as in the standard group plant, the reloading machine being of the traversing, instead of pivotal type. In the plant shown by Fig. 9 two of these machines are employed—one on each side of the storage area—and deliver to flight conveyors running in and towards the middle of an underground tunnel which is laid parallel to the traverse of the trimmer. From these flight conveyors the coal is delivered to a bucket conveyor, placed at right angles to them, which carries to a reloading tower for car delivery.

10 Application of the anthracite storage machinery covers successfully the requirements of water-front delivery. At the plant shown by Fig. 10 the coal is received altogether by boat and is unloaded by means of automatic buckets, which serve a conveyor of large capacity, terminating in a central tower, from which chutes distribute to the trimmer conveyors. Reloading is similar to that of the inland storage plants.

COVERED ANTHRACITE STORAGE

11 Where climatic severity makes covering of the coal necessary the "Dodge System" has been found especially adapted to the conditions governing location and transportation facilities. A prominent installation of this kind employs a circular building with trussed dome-shaped roof. The coal is brought to the dock by boat, and unloaded by steeple tower hoist, from which it is spouted by gravity to the chute of the trimmer conveyor. The construction of the trimmer permits it to be mounted on one of the roof trusses of the building. The reloading from pile is effected by means of the pivotal reloader which commands the inner circumference of the building, and delivers through floor openings to a flighted chain conveyor running, horizontally in a tunnel, from a position beneath the center of the pile to a point outside the building. From here it rises on an incline to the reloading tower from which its discharge is transferred through screening chutes to cars. No exterior bracing is required for these buildings, as they are strengthened against bursting pressure from the mass of coal by a continuous circular bulk-head which is supported free of the floor by anchor bolts extending inwardly from the I-beam posts of the annular wall.

12 Another type of covered storage plant is shown by Fig. 11, in which the housed area is of rectangular form, the building being

Fig. 11 Type of Covered Storage Plant

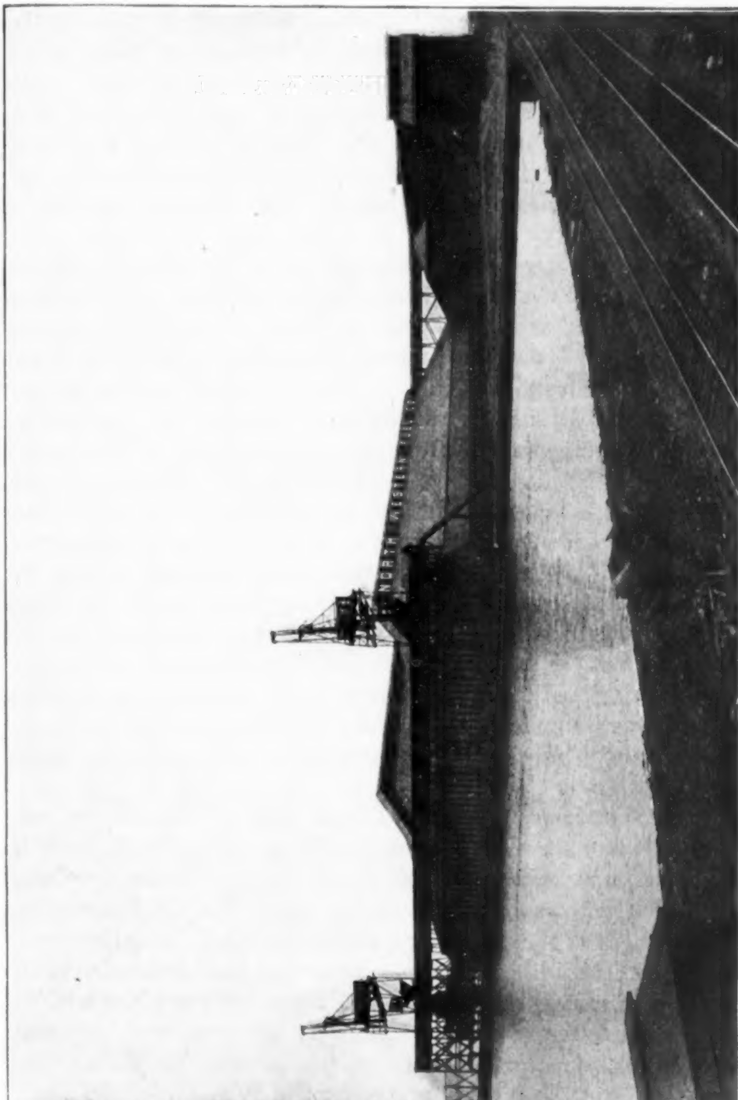

FIG. 12 WATER FRONT COVERED STORAGE PLANT

divided to store equal quantities of egg, stove, and nut coal, the aggregate capacity footing up to 100 000 tons. The equipment consists of three standard "Dodge" trimming conveyors for stocking the coal; three open-top carriers, running in underground tunnels, for reloading; three reloading towers provided with screens; three gravity-discharge transfer conveyors; and three screenings conveyors. The three trimmer conveyors and three transfer conveyors, or three reloading conveyors and three transfer conveyors, can be operated simultaneously. Each machine handles 1500 net tons per day of ten hours.

13 Another water-front storage plant of the covered type is illustrated by Fig. 12 and 13. Here the coal is unloaded from vessels by steeple towers, and discharged into a wharf conveyor, carrying on its upper and lower run, and which transfers to reversible flight conveyors for delivery to storage building. Movable ribbon troughs permit gradual discharge and eliminate breakage. Direct re-shipment from storage, or from the storage conveyors, is effected by another reversible flight conveyor which serves a shipping pocket. To reload from storage, the desired size of coal may be drawn from underneath the pile through chutes to conveyors running lengthwise of the building, and which discharge into the conveyor serving the shipping pocket. Coal tributary to this conveyor may be fed direct to it. From the shipping pocket coal is conveyed to box cars served by a Smith box car loader. Screenings are automatically discharged to a conveying and elevating system which commands a 3000 ton screenings storage building. From this point they are sized and loaded into box cars in the same manner as from the size-coal shipping pocket.

14 The power plant for a "Dodge System" installation may include steam or electrical equipment, the generating outfit comprising a central unit from which distribution may be effected through a piping or wiring system to convenient points for direct operation. The equipment of the large plant, shown by Fig. 7, includes four 16 in. by 20 in. 150 h.p. engines so located that one engine drives the machinery of each two-pile group. In a 240 000-ton plant which is nearing completion, two 225 h.p. alternating current, 440-volt, 25-cycle, 3-phase motors drive the machinery of the 60 000-ton piles, while that of the 30 000 ton piles is controlled by motors of 150 h.p.

15 The capacity of the plants erected under the "Dodge System" aggregates a total of about 5 000 000 tons. While official figures are not obtainable for publication, it is guaranteed that the labor cost of

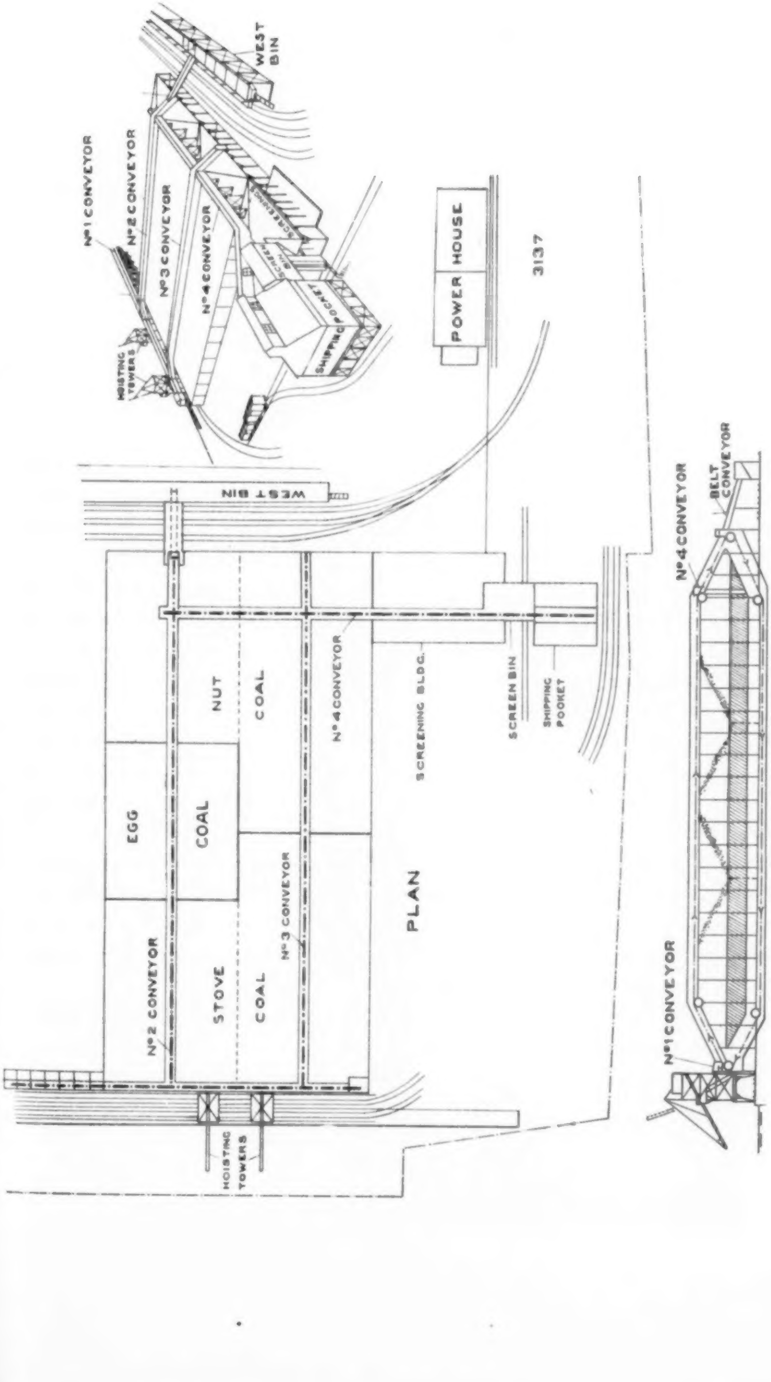

Fig. 13 Plan and Section of Plant Shown in Fig. 12

either stocking out or reloading on an active plant will not exceed four cents per ton.

BITUMINOUS COAL STORAGE MACHINERY

Bituminous coal is generally handled by three methods.

16 (*a*) A Robins Conveying Belt Co.'s stocking and reclaiming machine, Fig. 14, consists of a lateral-traveling bridge tramway spanning the storage area, the coal being supplied by a feeding conveyor running parallel to the storage area, and delivered to the storage pile by a belt conveyor equipped with an automatic tripper. The coal is reclaimed by a counterweighted self-filling bucket and delivered to the longitudinal feeding conveyor for transmission to the point of consumption.

17 (*b*) Another type of machine used for this purpose is the J. M. Dodge Company rotating bridge tramway shown by Fig. 15. This machine has a span of 280 ft., and commands a circular storage pile of 100 000 tons, the coal being piled 30 ft. in depth. The inner end of the bridge is pivoted and the outer end is supported on structural steel legs resting on power-driven trucks, which travel on a semi-circular track at the rate of 200 ft. per min. The tramway is equipped with a self-filling bucket of four tons capacity, and the entire operation of the bridge is controlled by one man. The coal is dumped from railroad cars into a track hopper pit located under the center of rotation of the bridge. The bucket, which travels on a track on the bridge, lifts the coal from this pit through the open center of the turntable at a speed of 300 ft. per min. and trolleys it at a speed of 1000 ft. per min. for discharge to the storage pile. The coal is reclaimed by the bucket, delivered to a receiving hopper placed under its path, and carried by means of conveyors or cars to the point of consumption. The machine is electrically operated, the bucket being hoisted by means of one 200 h.p. motor and trolleyed by one 100 h.p. motor. Rotation is effected by the trucks located on the outer legs and operated by two motors of 30 h.p. each.

18 The cost of handling coal for the first nine months of 1906 with this machine was as follows:

FIG. 14 ROBINS STOCKING AND RECLAIMING MACHINE

FIG. 15 THE J. M. DODGE COMPANY'S ROTATING BRIDGE TRAMWAY

Month	Hours in operation	Tons	Renewals	
			Labor	Material
Jan.	100 at 43 cents	5 210	$43.59	$135.43
Feb.	80 at 43 cents	4 740	64.86	0.00
Mar.	100 at 43 cents	6 210	52.87	149.51
April	210 at 43 cents	13 800	35.83	35.36
May	130 at 43 cents	7 710	65.56	171.06
June	36 at 43 cents	1 710	55.75	0.00
July	204 at 43 cents	12 340	26.36	24.22
Aug.	148 at 43 cents	7 980	57.27	223.64
Sept.	71 at 43 cents	3 780	50.87	0.00

Summarized, this gives the following:

Labor cost, handling coal	$0.0073 per ton.
Labor cost for renewals	0.0069 " "
Cost of renewals in material	0.0116 " "
Total	0.0258 " "

19 (*c*) For storing from 6000 to 40 000 tons of coal a long-radius J. M. Dodge Company locomotive crane is used. This machine runs on a circular track around a central track-hopper into which the coal is dumped from railroad cars, as shown by Fig. 16. The coal is taken from this pit by a self-filling bucket and delivered to the storage pile, which when full assumes the outline shown in the plan. (See Fig. 17.) When reloading, the coal is taken from the pile by the bucket, and delivered into cars or carts.

STORAGE FOR LOCOMOTIVE COALING STATIONS

20 Railroads also employ this type of crane for handling and storing coal for their locomotive coaling stations. These machines are steam-operated, and are equipped with a 2-ton self-filling bucket operated at 80 ft. radius. The coal is taken from the track-hoppers and delivered to storage, and from storage to a pocket fitted with chutes for coaling six locomotives simultaneously.

The cost of operating the plant referred to for the month of December, 1907, was as follows:

Total cost of labor for unloading cars, operating cranes, and switching service, also repairs, fuel, and supplies,	$451.96
Total tons of coal in storage on first day of month	38 717
Tons of coal stored during the month	725
Cost to handle per ton	$0.0529

Fig. 16 Coal Storage Locomotive Crane

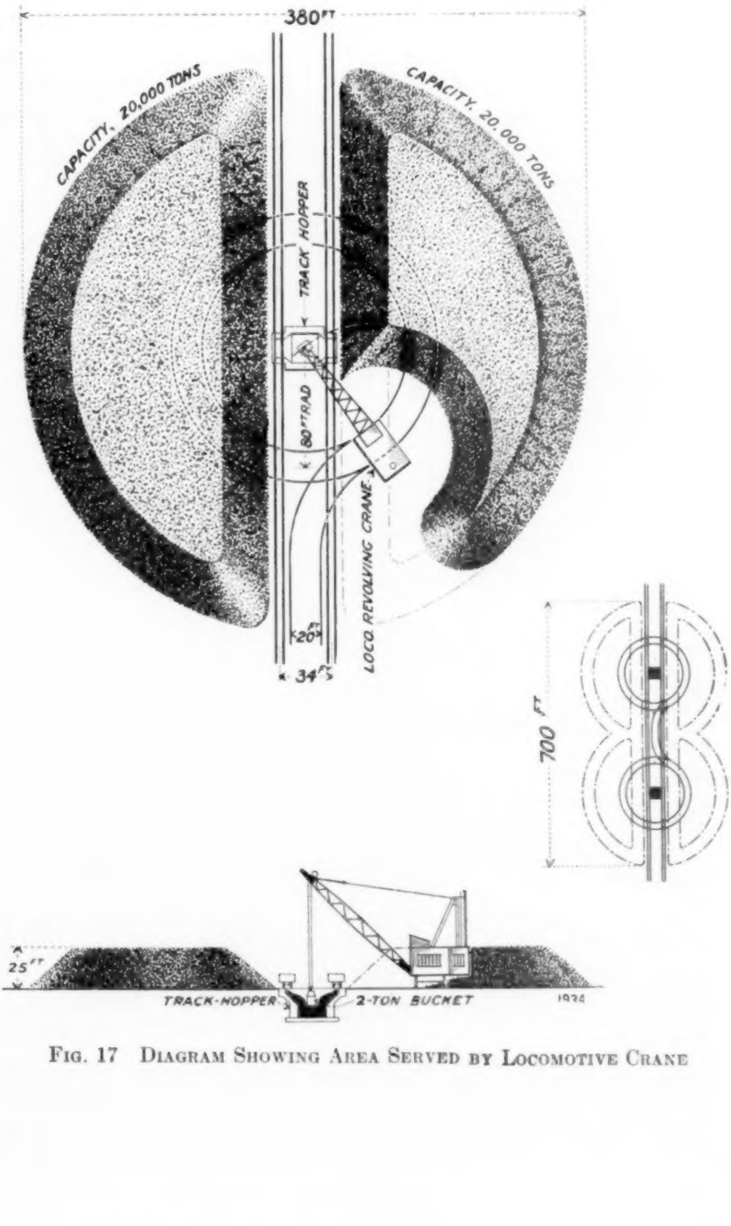

FIG. 17 DIAGRAM SHOWING AREA SERVED BY LOCOMOTIVE CRANE

Tons of coal delivered to engines,	9624
Cost per ton to handle,	$0.0429
Tons of coal reloaded for shipment,	none
Tons of coal in storage on last day of month,	35 574
Tons of coal handled during the month,	10 349
Average cost per ton to handle,	$0.0437
Number of hours operated,	210
Tons of coal handled per hour,	49.28

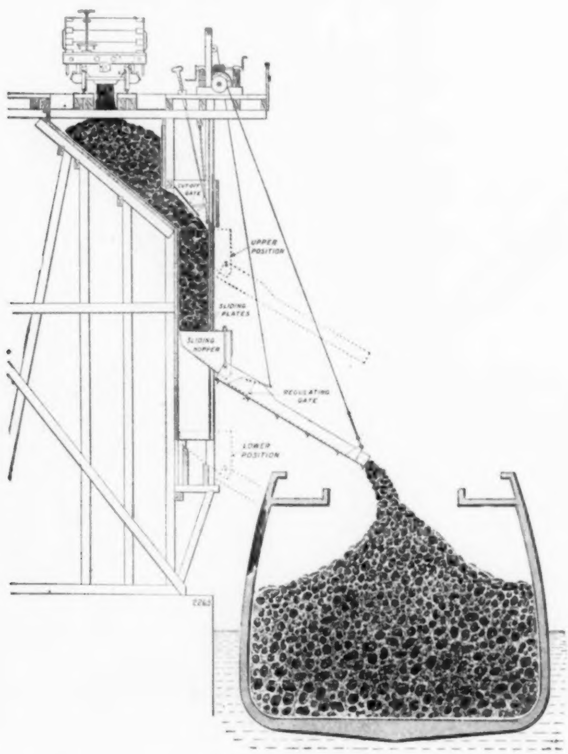

FIG. 18 ADJUSTABLE CHUTE FOR LOADING VESSELS

MACHINERY FOR LOADING COAL INTO VESSELS

21 On the Atlantic seaboard bituminous and anthracite coal is loaded in a large number of instances through the same type of pocket as that used for loading iron ore at Lake Superior ports, except that special devices, as shown by Fig. 18, are provided for reducing the breakage of coal between the car and the vessel. The

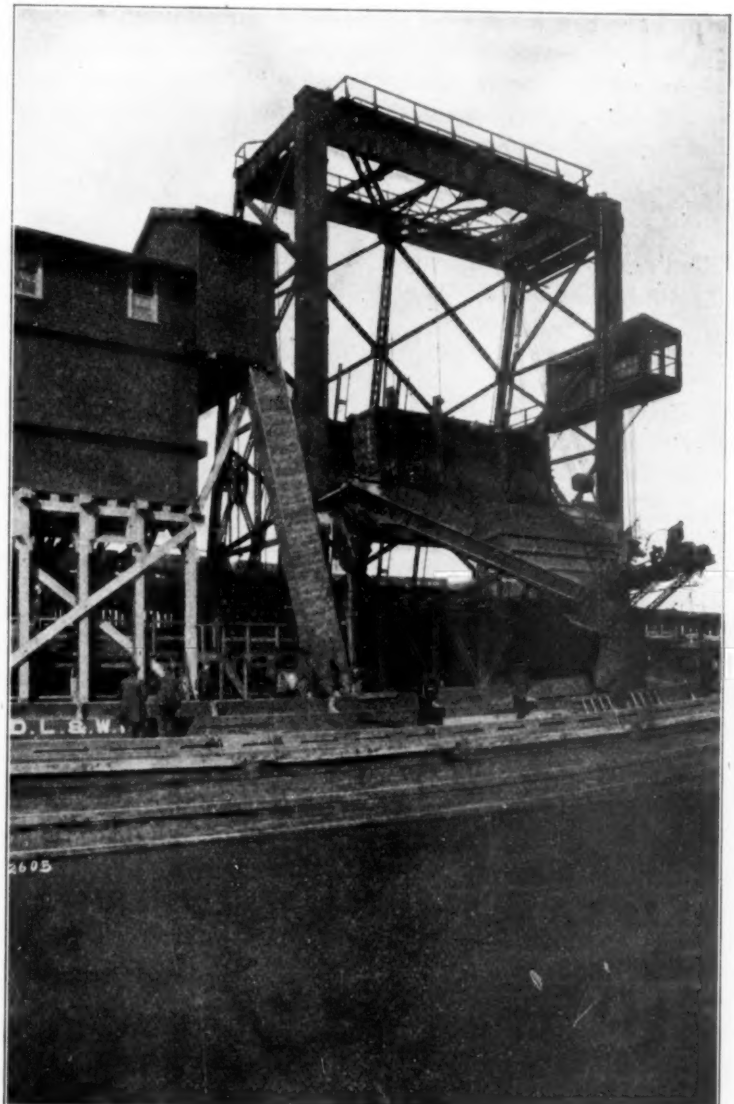

FIG. 19 CAR UNLOADING MACHINE

coal coming from the Eastern and Middle States in cars to shipping ports on Lake Erie is delivered to vessels by car unloading machines similar in design to that shown by Fig. 19. This machine is arranged to take cars of any size or capacity found in the trade, elevate, overturn, and deliver the contents through a telescopic chute into the hold of the vessel, and return the cars to the railroad system.

Maximum number of cars handled in one hour, 32.
" " " tons " " " " 1062.
Best record of loading vessel "Zenith City," 130 cars, or 4240 tons, in 4½ hours. Average, 29 cars, or 942 tons per hour.

MACHINERY FOR UNLOADING COAL FROM VESSELS

22 The coal on arrival at the upper lake ports is unloaded in a variety of ways, depending upon the character of the coal, the purpose for which it is intended, and the local conditions. A typical plant constructed by The Mead-Morrison Mfg. Co. is shown by Fig. 20. This plant consists of a rectangular storage wharf, on the water side of which, traveling on an elevated trestle, are three unloading machines equipped with self-filling buckets. Coal is unloaded by these machines and automatically delivered to a system of elevated cable cars running under the unloading towers and over continuous shipping pockets located on the center line of the property; or, if the coal is intended for storage, they are diverted to a traveling bridge and discharged onto the storage area. To reload the coal a shovel-bucket, with which the traveling bridge is equipped, is used to deliver directly into the pockets or to the cable cars for discharge along the line of the pockets.

23 Another type—the J. M. Dodge coal storage plant shown by Fig. 21. This plant is designed to avoid breakage as much as possible, and consists of three traversing bridge tramways commanding a storage of 250 000 tons of bituminous coal. The bridges are supplemented by two revolving locomotive cranes, two movable screening towers, and two box car loaders. The coal is unloaded from vessels by self-filling buckets of 3-tons capacity, and is delivered directly to storage, or through reloading hoppers, on front tower of tramways to cars for reshipment. Transfer from storage to cars is accomplished by similar bucket delivery to the reloading hoppers.

24 When size-coal shipments are desired, coal is taken from vessels or storage by bridge tramway or crane to the movable screening towers,

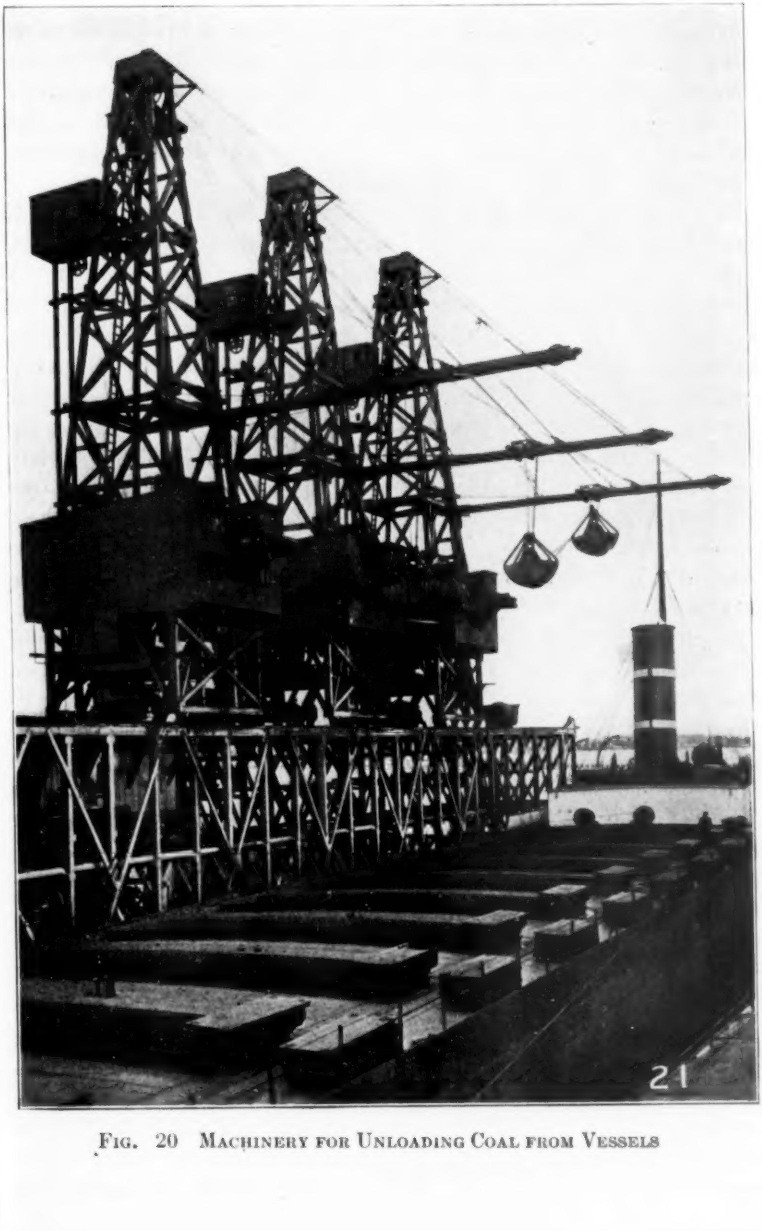

FIG. 20 MACHINERY FOR UNLOADING COAL FROM VESSELS

from which it is delivered to box cars alongside. Slack coal screened from size-coal shipments is automatically discharged into cars, or conveyed to slack storage, from which the cranes deliver to the movable screening towers for box car delivery as needed. Delivery to box cars, as noted, is facilitated in all cases by box-car loading machines of the Ottumwa type, operated by 100 h.p. motors. All of the machinery is controlled and operated by means of alternating current apparatus.

25 A novel distributing system delivers the 3-phase, 440-volt, 25-cycle alternating current by means of a series of contact posts, which in no way obstruct free passage of men or of material on the wharf. The bridge tramways have a total length of 506 ft., composed of a main span of 295 ft., an adjustable boom extension of 75 ft. over the vessel, and a rear cantilever extension of 136 ft. The electrical equipment of each tramway consists of one 225 h.p. motor for operating the hoisting machine, and two 75 h.p. motors for trolleying the bucket and moving the bridge along the dock. The complete operation of each bridge is controlled by one man from either of two cabs—one located at front of bridge and nearly over the hatch of the vessel, and the other adjacent to the rear tower. In this way the operator's view is unobstructed whether working at the front or the back of the wharf.

26 A comprehensive description of the screening towers would be to compare them to traveling mine tipples. Each tower is equipped with a 30-ton hopper for receiving run-of-mine coal from the self-filling buckets on the tramways or cranes. The hoppers are provided with standard screen bars, the spacing of which can be changed at will. An elevator and conveyor carry the screenings across the screening-tower bridge and deposit them on screening pile at the back end of the wharf. The screened coal is loaded into box cars by the electrically-operated box car loaders referred to previously. The cranes have a gage of 10 ft., and a maximum radius—from the center of rotation to the center of the bucket—of 45 ft. The traveling speed of each crane is 300 ft. per min., driven by 75 h.p. motor, and the hoisting speed, 250 ft. per min.

27 This plant has the following record of performance: During the month of October 1907, the three bridge tramways unloaded from vessels and discharged 85 266 tons in 239 hours. In handling this tonnage the buckets made 34 933 trips, one-half the coal being lowered very carefully to the wharf. A round trip from the center of the vessel over a traveled distance of 270 ft. and return was made in 45

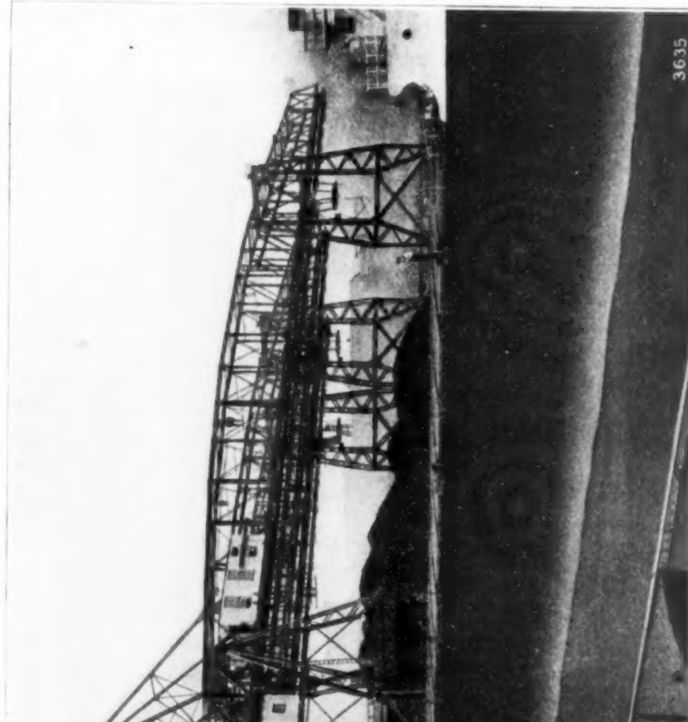

FIG. 21 BRIDGE TRAMWAY TYPE OF BITUMINOUS COAL STORAGE PLANT

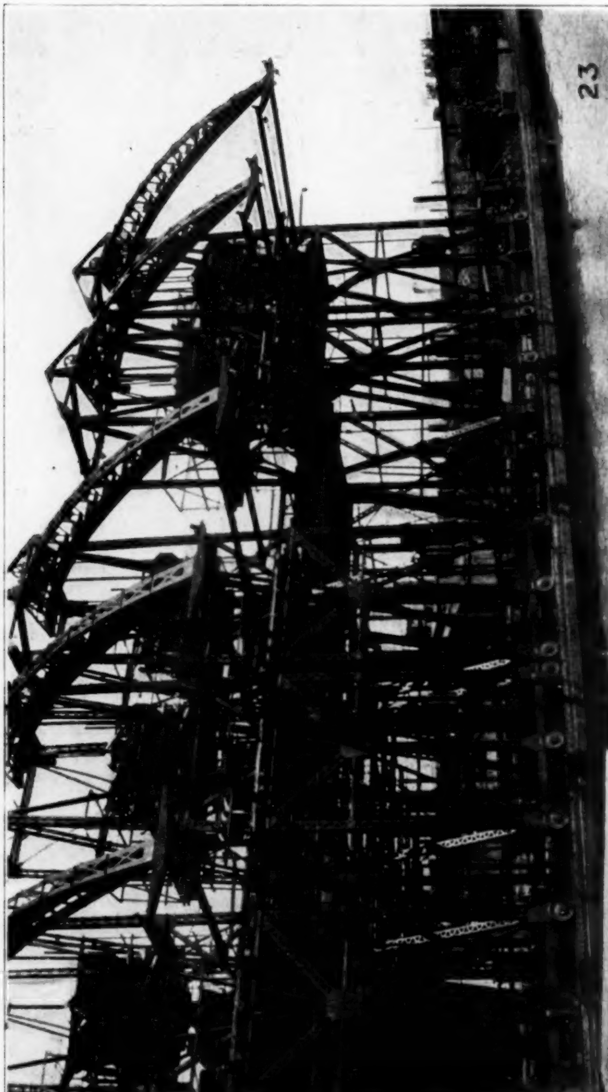

Fig. 22 Parabolic Boom Unloading Towers

seconds. In one hour one bridge made 100 round trips from vessel to front of storage pile. The three bridges made 822 trips in 693 minutes, and disposed of a cargo of 10 500 tons in 26 hours.

28 Another type of coal unloader, as shown by Fig. 22, is made by the C. W. Hunt Co. This plant has five unloading towers fitted with parabolic booms which handle two-ton shovels. There are 27 automatic railway tracks to carry the coal from the storage sheds.

VARIATIONS IN DESIGN—FREIGHT HANDLING

29 Machines similar to those in use for unloading ore at Lake Erie ports are also suitable for handling general merchandise between cars and ships. The J. M. Dodge Company, direct unloading machine shown by Fig. 23 has two independent tracks, one of which is equipped with hoisting mechanism, for handling general merchandise, and the other with a self-filling bucket for handling bulk material. This machine is electrically-operated by means of alternating current.

The hoisting speed of a 10-ton load is	150	ft. per min.
" " " " 5-ton " "	300	" " "
The hoisting speed of 1-ton clam-shell bucket is	300	" " "
The trolleying speed of any load is	500	" " "
The traversing speed of unloader is	50	" " "

The control of the motors is such that any load from an empty hanging-block to the maximum load of ten tons, can be operated at any required speed.

30 Fig. 24 shows a man-trolley similar to those used for unloading iron ore from vessels, the construction of the trolley being such that it passes freely around curves and over switches. These machines are used for handling materials at blast furnaces, and freight in large industrial plants. They may have a hoisting speed of 150 ft. per min., and a trolley speed of 1000 ft. per min. The man-trolley type of machine also operates on cableways of long span. The towers of the cableway are inclined to the horizontal, and provided with counterweights. The self-filling bucket, used in all cases where it is desirable to handle the material with the least possible hand labor, has been applied to traveling cableways.

31 Another type of apparatus for handling and transporting materials of various kinds is the familiar suspended cableway. The

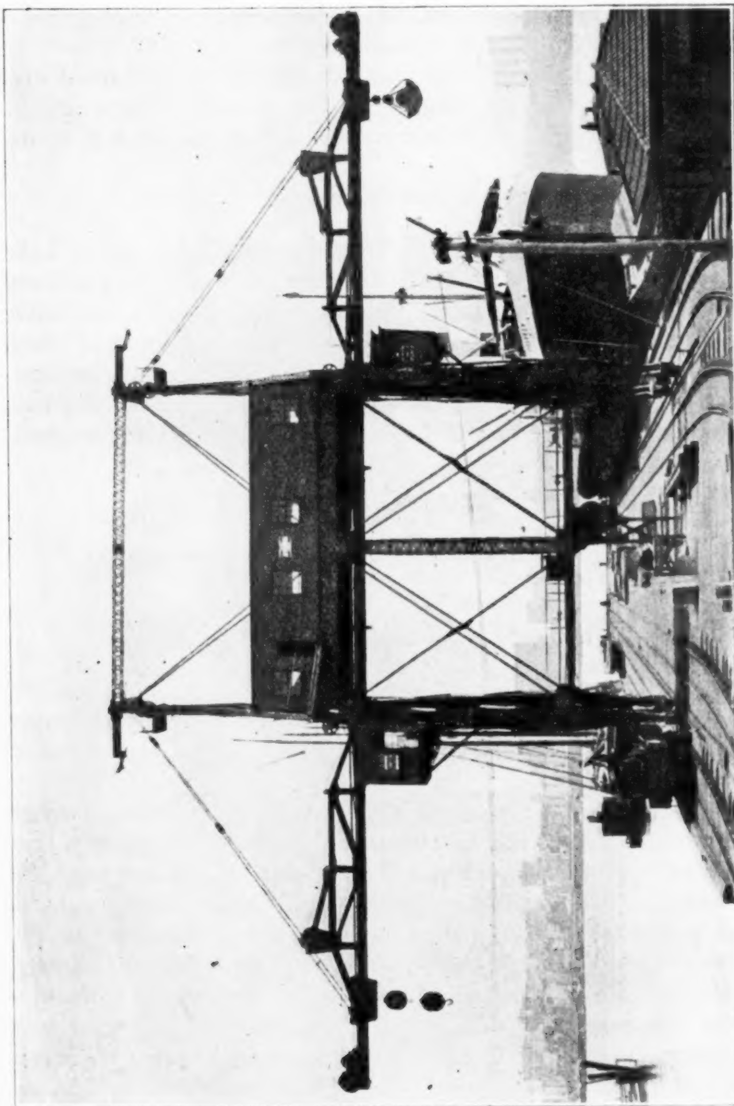

Fig. 23 Direct Unloader for Handling Freight

FIG. 24 MAN-TROLLEY FOR HANDLING FREIGHT

use of suspended cables for supporting the bucket makes it possible to construct machines of long span economically.

32 For installations where the quantities to be handled are not great, revolving locomotive cranes with self-filling buckets are used to take the material from vessels or cars, and deliver it into storage or cars.

33 The following gives the performance of two Dodge Coal Storage Co., standard-gage revolving locomotive electric cranes:

Material handled—crushed rock, $\frac{3}{4}$ in. to 2 in. mixed.

Bucket, 1 cu. yd., clam-shell.

Radius of crane, 35 ft.

Size of motor, 50 h.p., d.c., 500 volts.

Crane No. 1, from car to pile: Average capacity for ten minutes, 16 buckets.

Crane No. 2, car to street railway car: Average capacity:—20 buckets in 20 minutes.

Maximum rate—two buckets in one minute.

Buckets filled to seven-eighths capacity.

NOTE—The discussion on this paper and the author's closure are published under "Discussion on the Conveying of Materials," No. 1195 in this volume.
—EDITOR

No. 1191

CONTINUOUS CONVEYING OF MATERIALS

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Member of the Society

CHAIN CONVEYORS

The manufacture of conveying machinery, and its adaptation to the various industries, dates from the year 1880, when a company was incorporated for manufacturing and developing such machinery. This company was organized by the manufacturers of the Ewart detachable link belt in response to the demand for these links with attachments to which pushers or buckets could be fastened, and to requests for advice about their utilization for all sorts of conveying purposes.

2 Although the development of this class of machinery began nearly 30 years ago, there is scarcely any literature on the subject outside of trade catalogues and comparatively few industries follow such uniform methods, or handle materials in such quantities, as to have developed standard designs for their conveying equipment. Figures upon economies effected have not been kept to the extent they should have been. Managers of industrial plants are generally satisfied with the apparent fact that there is a material saving and do not take the trouble to ascertain just how much this is, or whether it might not be greatly increased by the use of conveyors of improved construction.

3 This lack of information has given first cost an altogether unwarrantable importance in determining the character of installations of this kind. The initial cost should always be given its due consideration, but should not be used as a cloak for ignorance. Unfortunately a very large proportion of conveying plants are, from the nature of their service, very dirty and so located that investigation is conducted with more or less discomfort, but this should not

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

deter engineers from such investigation. The infinite variety of physical characteristics presented by materials handled indicates in itself the likelihood of wide variations in the designs of conveying machin-

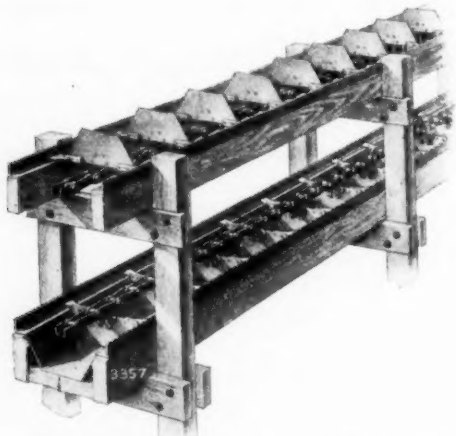


FIG. 1 CONVEYOR WITH SLIDING WEARING SHOES

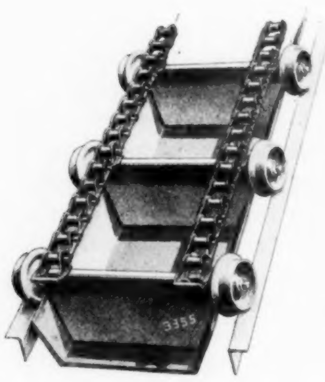


FIG. 2

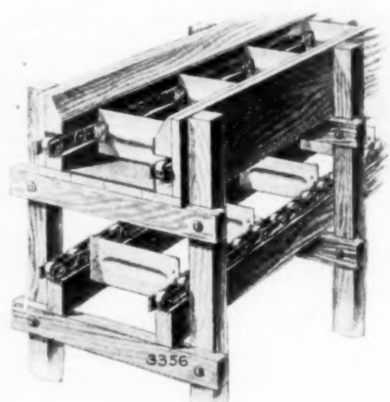


FIG. 3

FIG 2 ROLLER FLIGHT CONVEYOR
FIG. 3 REVERSIBLE FLIGHT CONVEYOR

ery to meet these conditions, and the necessity for careful examination of both design and details of construction by the engineer who

is called upon to decide between different plans that may be submitted to him.

4 The development of the conveying art has been greatly assisted by the industrial consolidations of the last seven or eight years. These have increased the raw and manufactured products of existing plants and have led to the building of new plants of such size that the economical handling of materials in process has wisely been considered from the outset and has had its determining influence on the plan as a whole. It is with such initial consideration, and in such plants, that the conveying art has reached its highest development and effected its greatest saving. In the discussion which follows some of

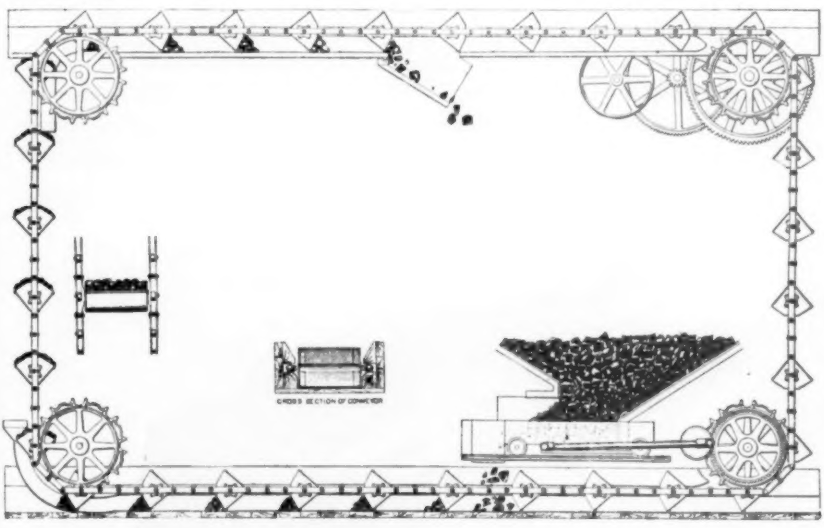


FIG. 4 BUCKET CONVEYOR WITH RIGID BUCKETS

the types of conveyors are considered which have proved most efficient and are in most general use.

5 Conveyors for the continuous handling of materials may be divided into two general classes:

- *a* Those which push or pull their load, the weight of the load not being borne by the moving parts of the conveyor.
- *b* Those which actually carry the material handled.

6 Conveyors of the first class push or pull the material handled along in a trough. The friction of the conveyor itself and the mate-

rial conveyed on the trough consume power and cause wear. Hence the field of usefulness of conveyors of this type is confined to relatively small conveyors with light service; or in the larger installations, to the handling of materials with a low coefficient of friction and which are not abrasive in their action, such as coal, grain, etc.

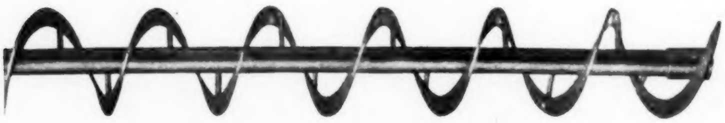


Fig. 5 Spiral Conveyor

7 One of the oldest forms, which, from its simplicity and comparatively low first cost is still one of the most extensively used, consists merely of an endless chain to which are attached at intervals scrapers or flights. The improved forms of this conveyor, now most generally used, have sliding shoes or rollers attached to the flights or the chains,

Fig. 6 Spiral Conveyor and Trough

supported on runways. The flights are allowed to come very close to the trough bottom, but not actually in contact with it, thus reducing the friction upon the trough to that of the material conveyed only. Fig. 1, 2, and 3 illustrate conveyors of this type, the rollers in Fig. 3 being an integral part of the chains.

TABLE 1 CONVEYING CAPACITIES OF FLIGHT CONVEYORS

in tons (2000 lb.) of coal per hour at 100 ft. per min.

SIZE OF FLIGHT	HORIZONTAL				INCLINED		
	Spaced			lb. per flight	10 deg. 24 in.	20 deg. 24 in.	30 deg. 24 in.
	16 in.	18 in.	24 in.				
4 by 10	33¾	30	22½	15	18	14½	10½
4 by 12	42¾	38	28½	19	24	18	13½
5 by 12	51¾	46	34½	23	28½	22½	16½
5 by 15	69¾	62	46½	31	40½	31½	22½
6 by 18		80	60	40	49½	40½	31½
8 by 18		120	90	60	72	57	48
8 by 20			105	70	84	66½	56
8 by 24			135	90	120	96	72
10 by 24			172½	115	150	120	90

8 The horse power required handling anthracite coal, may be determined from the following formula; this taking no account of gearing or other driving connections.

$$H.p. = \frac{ATL + BWS}{1000}$$

T = net tons per hour.
L = length center to center in feet.
W = weight, chain and flights (both runs) in pounds.
S = speed per minute in feet.

A and B are constants depending on inclination from horizontal, taken from Table 2.

TABLE 2

	Hor.	5 deg.	10 deg.	15 deg.	20 deg.	25 deg.	30 deg.	35 deg.	40 deg.	45 deg.
A	0.343	0.42	0.50	0.585	0.66	0.73	0.79	0.85	0.90	0.945
B	0.01	0.01	0.01	0.01	0.009	0.009	0.009	0.008	0.008	0.007

9 The common working speeds are from 100 ft. to 200 ft. per min. and the capacities are as shown by the following table, these conveyors in some cases handling upwards of five hundred tons per hour. Fig. 4 shows a modification of a conveyor of the general construction of Fig. 3, in which the flights are given the form of buckets. With this type the material will fall back into the buckets when the conveyor makes an upward turn and thus be elevated. On the turning of the conveyor again into a horizontal path, Fig. 4, the material partially flows out into the horizontal trough, whence it may be conveyed to the desired point of discharge. As this device combines in one machine the ability to handle material horizontally and vertically,

it has a wide field of usefulness, limited by the disadvantages already referred to as incident to all conveyors drawing material along in a trough.

TABLE 3 CAPACITIES OF GRAVITY DISCHARGE ELEVATOR CONVEYORS

in tons of coal per hour at 100 ft. per min.

BUCKET	SPACING			
Length and width in inches	18 in.	24 in.	36 in.	48 in.
12 by 12	31.0	23.5	15.5	11.7
16 by 12	41.3	31.3	20.5	15.6
16 by 15	61.4	46.5	30.7	23.3
20 by 15	78.0	59.0	39.0	29.5
24 by 15	91.0	69.0	45.5	34.5
20 by 20	138.0		69.0	
24 by 20	165.0		82.5	
30 by 20	206.0		103.0	
36 by 20	248.0		124.0	
30 by 24		220.0	145.0	110.0
36 by 24		262.0	173.0	131.0
48 by 24		350.0	232.0	175.0

TABLE 4 HORSE POWER OF ELEVATOR CONVEYORS

SIZE OF BUCKET	12 by 12 in.			24 by 15 in.			36 by 20 in.		48 by 24 in.	
Spacing	18 in.	24 in.	36 in.	18 in.	24 in.	36 in.	18 in.	24 in	36 in.	48in.
H.p. for each 10 ft. vert. lift	0.46	0.35	0.23	1.35	1.0	0.67	3.8	1.9	3.5	2.7
H.p. for each 100 ft. hor. empty	1.2	1.1	0.9	1.7	1.5	1.3	2.4	1.6		
Do handling anthracite	2.5	2.1	1.5	5.3	4.3	3.1	12.4	6.6		
Do handling bituminous	3.2	2.6	1.9	7.4	5.8	4.2	18	9.4		

Add 5 per cent for each turn.

SPIRAL CONVEYOR

10 The screw or spiral conveyor has a great advantage, aside from relatively low first cost, in its compactness and the small space it

occupies, adapting it to conditions where the conveyors have to be placed in a trench beneath the floor or where the maintenance of head room is an object. The conveyor consists of a steel ribbon wound spirally about a shaft revolving in a trough which it fits approximately, the spiral being interrupted for a space of two or three inches at intervals of eight or ten feet to permit the support of the shaft by hangers. In the smaller diameters—up to 12 in. or 16 in.—the spiral fits close to the shaft. In the larger sizes, running up to as much as 3 ft. or more in diameter, the spiral is a comparatively narrow ribbon attached to the shaft by occasional studs. (Fig. 5.) The depth of material that it is practicable to convey is approximately one-third of the diameter of the screw.

11 Spiral conveyors, like all scraper conveyors, are costly of maintenance when handling gritty materials, and in this case heavy cast iron sectional and renewable flights are used running in a cast iron trough with hangers having white iron bushings. It is considered good practice under these conditions to make the box in which the conveyor runs of a size considerably larger than the conveyor, letting the material form its own trough. On account of the torsional strain on the shaft, screw conveyors are not very generally used for lengths exceeding about 100 ft., though where the service is exceptionally easy like the handling of grain or cotton seed, this length may be greatly exceeded. The following table gives a fair idea of the conveying capacities:

TABLE 5 CAPACITIES OF SPIRAL CONVEYORS IN CUBIC FEET OF MATERIAL CONVEYED PER HOUR, AT 100 R.P.M.

Diam.	6 in.	8 in.	10 in.	12 in.	14 in.	16 in.	18 in.
Cap.	260	660	1000	1660	2500	3660	5000

$$\text{H. p.} = \frac{\text{Capacity in lb. per min. times length in ft. by C}}{33\,000}$$

$$\text{C is a constant and equals} \begin{cases} 0.67 \text{ for coal} \\ 0.33 \text{ for grain} \\ 1.00 \text{ for ashes} \end{cases}$$

12 Still another conveyor, which operates without the aid of a chain, and like the screw conveyor has the advantage of occupying but little space, is shown by Fig. 7 and 8. The flights are here attached by hinged connection to a bar or frame to which is given a reciprocating motion. On the forward stroke the flights push the

material ahead, and on the backward fold up over it by means of the hinge. The moving frame is carried by rollers and the stroke or amount of forward travel is usually from 2 ft. to 4 ft. This conveyor has been used principally in the handling of sand and materials that would cause too much wear of a screw conveyor.

13 The foregoing conveyors handle for the most part material in bulk. Conveyors of this class, where the flight takes the form of a projecting spur, pusher or hook, are widely used for log hauls and car hauls, and also to some extent for packages and boxes, either horizontally or at an inclination. Such conveyors put in beneath the floor and with the pushers projecting through a narrow slot in the floor have been very serviceable in the handling of trucks in warehouses and manufacturing establishments.

FIG. 7 RECIPROCATING CONVEYOR

BELT CONVEYORS

14 The general class of conveyors designated in the beginning as Class *b* which carry their load, have the more universal adaptation to varying requirements. Of these the conveying belt is one of the oldest and still most useful. To a limited extent, and for very light service, the belt may be supported by a trough; but almost universally it is carried by rollers spaced on the carrying run from 2 ft. 6 in. to 5 ft. apart, and at intervals usually twice as great or more on the idle run. The earliest conveying belts were perfectly flat, being supported by plain cylindrical rollers such as are still used for the returning run. In order to increase the conveying capacity without the material spilling off the edges of the belt, the rollers were somewhat dished, or made concave in form, causing the belt to assume the form

of a shallow trough. In theory this is open to the criticism that but one diameter of the concave roller can travel at the speed of the belt, and some slipping must, therefore, take place at every other diameter of the periphery. Careful observations have shown, however, that the belts invariably fail in other ways before any injury from this slipping becomes apparent.

15 Before this fact was demonstrated, however, a supporting device, consisting of three independent rollers, the middle one horizontal and the side rollers inclined some 35 deg. or 40 deg. came into very general use, this arrangement giving the belt the form of a deep trough, and adding greatly to its carrying capacity.

16 Further modification of this was the substitution of two inclined center idlers for the one horizontal idler. This, while giving a relatively deep trough, overcame the sharp bends in the belt incident

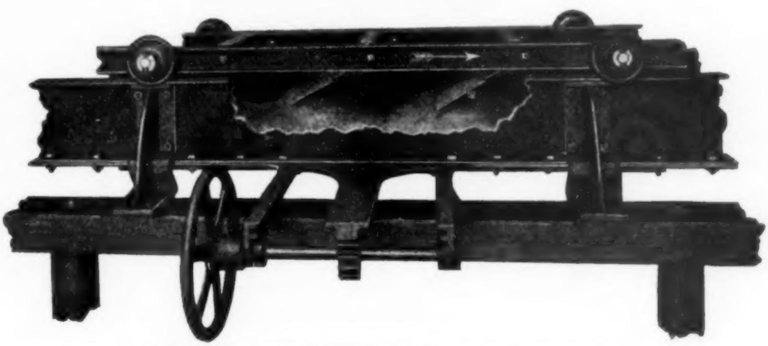


Fig. 8 Return Movement, Flights Deflected

to the three roll support, and permitted the belt to assume a uniform curve. For belts of large capacity the four roll idler may, therefore, be considered the best modern practice. For smaller belts and moderate capacities, there is nothing better than the old concave roller referred to, which troughs the belt but slightly, and, therefore, insures the greatest durability.

17 The conveying belts themselves are of cotton duck, woven solid; or of a number of plies varying from three to eight, stitched or cemented together with a composition of rubber, and known as rubber belts. Canvas belts are plain duck, or treated with some preservative and painted with similar compound. For many kinds of service they meet every requirement. For severer duty, where the cotton fabric, which is the strength of the belt, must be protected as perfectly

as possible from dust, moisture, and cutting or wearing action, the rubber belts are preferable, and are usually made with a cushion of from $\frac{1}{16}$ in. to $\frac{1}{4}$ in. more or less pure rubber on the carrying side, which protects the fabric until this cushion is worn away. Special types of belts have been extensively used, some having fewer plies of canvas and a heavier cushion of rubber in the center where the belt is designed to receive and carry its heaviest load, and others having the fabric made thinner at the points where it is intended the belt should be bent to form a trough. Experience seems to show that the greatest durability is attained by avoiding a localized bending.

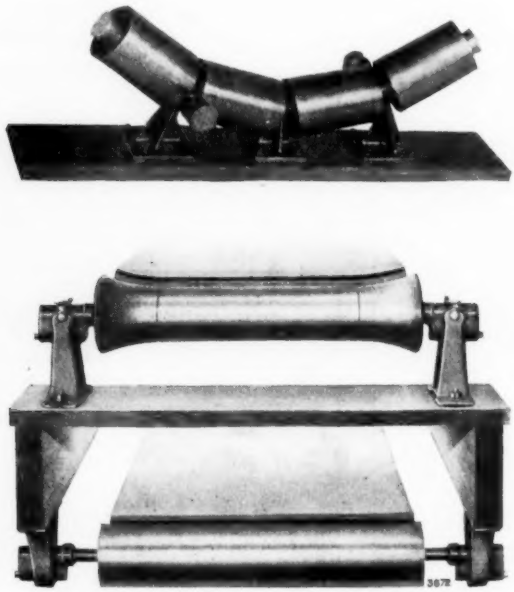


Fig. 9 Recent Types of Roller Support

18 The belt conveyor has a wide field of usefulness and is deservedly popular both with manufacturer and user. It is simple, smooth and noiseless in operation, may be run at relatively high speeds, from 300 ft. to 800 ft. per minute—with consequent large conveying capacity. On account of the expense of the belt, and the large number of supporting rollers which revolve at high speed, the initial cost and the power consumed in operation are much greater than would be sup-

TABLE 6 POWER REQUIRED FOR BELT CONVEYORS

FIGURES GIVEN ARE FOR CONVEYORS 100 FT. LONG, RUNNING 100 FT. PER MINUTE

Inclination of belt	HORSE POWER WITH DEEP TROUGHING ROLLS										HORSE POWER WITH SHALLOW TROUGHING ROLLS									
	Wt. of material 50 lb. per cu. ft.																			
	12	14	16	18	20	24	30	36	40	42	12	14	16	18	20	24	30	36	40	42
0 deg.	25	0.31	0.40	0.47	0.54	0.75	1.18	1.49	1.73	1.83	0.22	0.26	0.30	0.34	0.39	0.58	0.77	1.00	1.18	1.35
5	0.32	0.41	0.54	0.66	0.78	1.12	1.76	2.34	2.80	3.01	0.28	0.34	0.41	0.48	0.57	0.84	1.19	1.61	1.97	2.33
10	0.39	0.51	0.68	0.85	1.02	1.48	2.34	3.19	3.87	4.19	0.34	0.42	0.52	0.62	0.75	1.10	1.61	2.21	2.76	3.31
15	0.46	0.61	0.83	1.04	1.26	1.84	2.92	4.04	4.94	5.37	0.40	0.50	0.63	0.76	0.93	1.36	2.03	2.81	3.55	4.29
20	0.53	0.71	0.97	1.23	1.50	2.21	3.50	8.49	6.01	6.55	0.46	0.58	0.74	0.90	1.11	1.62	2.45	3.41	4.34	5.27
	Wt. of material 100 lbs. per cu. ft.																			
0 deg.	0.31	0.37	0.50	0.61	0.71	1.01	1.60	2.09	2.49	2.66	0.26	0.32	0.38	0.45	0.52	0.77	1.07	1.41	1.74	2.04
5	0.45	0.60	0.78	0.99	1.19	1.75	2.76	3.79	4.63	5.02	0.38	0.48	0.60	0.73	0.88	1.29	1.91	2.61	3.32	4.00
10	0.59	0.83	1.06	1.37	1.67	2.49	3.92	5.49	6.77	7.38	0.50	0.64	0.82	1.01	1.24	1.81	2.75	3.81	4.90	5.96
15	0.73	1.06	1.34	1.75	2.15	3.23	5.08	7.19	8.91	9.74	0.62	0.80	1.04	1.29	1.60	2.33	3.59	5.01	6.48	7.92
20	0.87	1.29	1.62	2.13	2.63	3.97	6.24	8.89	11.05	12.10	0.74	0.96	1.26	1.57	1.96	2.85	4.43	6.21	8.06	9.88

Add to the horse power for drive and head shaft 10 per cent if no discharger is used; 33 per cent with discharger; and 10 per cent for each 180 deg. idler.

TABLE 7 CONVEYING CAPACITIES FOR BELT CONVEYORS
at 100 feet per minute

Capacities with deep troughing rolls

Width of belt, inches	12	14	16	18	20	24	30	36	40	42
Capacity in cu. ft. per hr	320	450	670	900	1140	1700	2700	4000	5000	5500
Capacities in bushels per hr	256	360	536	720	912	1360	2160	3200	4000	4400
Capacity in tons per hour: 40 lb. per cu. ft.	6.4	9.0	13.4	18.0	22.8	34.0	54.0	80	100	110
50 " " " "	8.0	11.2	16.8	22.5	28.5	42.5	67.5	100	125	138
60 " " " "	9.6	13.5	20.1	27.0	34.2	51.0	81.0	120	150	165
80 " " " "	12.8	18.0	26.8	36.0	45.6	68.0	108.0	160	200	220
100 " " " "	16.0	22.4	33.6	45.0	57.0	85.0	135.0	200	250	275
150 " " " "	24.0	33.6	50.4	67.5	85.5	128.0	203.0	300	375	413

Capacities with shallow troughing rolls.

Width of belt, inches	12	14	16	18	20	24	30	36	40	42
Capacity in cu. ft. per hr	280	390	520	670	840	1240	1950	2800	3700	4600
Capacities in bushels per hr	224	316	416	536	672	992	1560	2240	2960	3680
Capacity in tons per hour: 40 lb. per cu. ft.	5.6	7.8	10.4	13.4	16.8	24.8	39.0	56.0	74.0	92
50 " " " "	7.0	9.8	13.0	16.8	21.0	31.0	48.8	70.0	92.5	115
60 " " " "	8.4	11.7	15.6	20.2	25.2	37.2	58.5	84.0	111.0	138
80 " " " "	11.2	15.6	20.8	26.8	33.6	49.6	78.0	112.0	148.0	184
100 " " " "	14.0	19.5	26.0	33.6	42.0	62.0	97.4	140.0	185.0	225
150 " " " "	21.0	29.3	39.0	50.4	63.0	93.0	146.4	210.0	277.0	344

TABLE 8 CAPACITY AND HORSE POWER OF PIVOTED BUCKET CARRIERS
Actual tests

Power House	Path	Speed per minute feet	H.P. REQUIRED RUNNING Light	Loaded	Tons per hour
A street railway	Horizontally 200 ft. vertically 110 ft. then horizontally 180 ft. and return	54	9.9	21.0	92.3
B railroad shops	Rectangular 64 ft. horizontally and 49 ft. vertically	45	2.7	4.6	33.0
C electric lighting company	Rectangular 152 ft. horizontally and 52 ft. vertically	45	3.3	5.0	30.40
D department store	Vertically 114 ft. horizontally 75 ft. and return	42	6.17	15.4	40.0
E navy yard	Rectangular 103 ft. horizontally 33 ft. vertically	46	3.0	5.0	30.0

Conveyor drives reciprocating feeder

posed, and not materially less than heavier and more cumbrous looking conveyors of other types performing equivalent service.

19 The most serious objection to belt conveyors, and the one which has prevented their even more general use, is the lack of durability of the belts, their liability to destruction from accidental causes, and the expense of their frequent renewal. Their average life probably does not exceed two to three years, in spite of the fact that there are many individual instances where they have given much longer service.

20 A further objection is the difficulty of delivering at intermediate points, which is accomplished through the medium of a rather

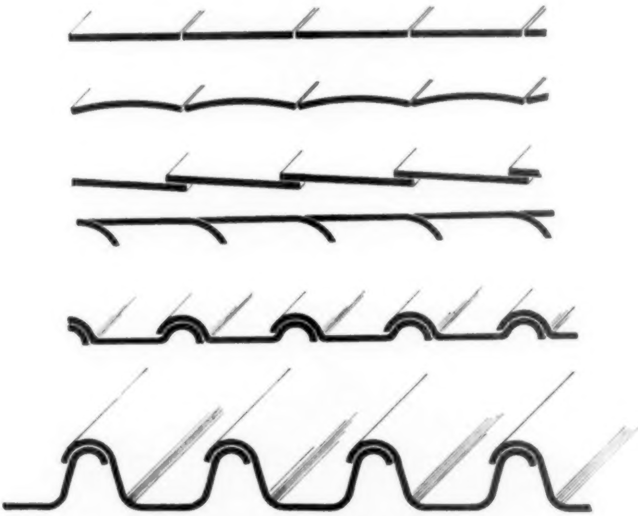


FIG. 10 SECTIONS SHOWING FORMS OF PLATFORM CONVEYORS

cumbrous and slow moving discharger. For filling long bins, however, a discharger is now used which travels slowly back and forth the length of the conveyor, its movement being automatically reversed at the ends of its path.

APRON AND BUCKET CONVEYORS

21 The most elementary form of link or chain belt conveyor approximating most nearly the belt, consists of flat steel or wooden slats attached between two chains, forming a continuous platform or

apron. The diagrams in Fig. 10 show successive modifications of this until the overlapping gravity bucket carrier is reached. Fig. 11 shows a plain, flat-top, steel apron. Where the slats are bent in arcs of the diameter of the head wheels, in passing around these wheels the scraper or chute may be placed close to the apron, thus

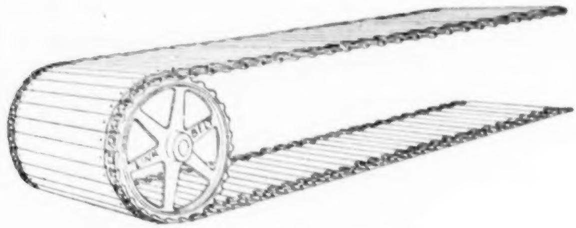


FIG. 11 ENDLESS APRON

adapting it to handle such sticky materials as clay, sugar, etc. In Fig. 12 the carrying slats are beaded at each edge, thus greatly stiffening them and increasing their ability to carry heavy loads, while the overlapping of the curved edges keeps the material from sifting through, making in this respect these carriers the equivalent

FIG. 12 BEADED SLATS

of a true belt. Fig. 13 shows this principle of construction carried still further, forming a corrugated apron, the elements of which are almost pan shape; and in Fig. 14 these have become real pans, and are adapted, as are the carriers shown by the two preceding figures, to carrying materials horizontally or at inclinations of 25 deg. to 30 deg., and to the heaviest loads.

22 None of these carriers are designed to elevate material except at an inclination, nor can they discharge except at the end of their run in passing around the sprocket wheels. Fig. 15 shows a still further evolution, this carrier, from the form of the pans, being enabled to convey on the horizontal run and also ascend vertically, delivering

FIG. 13 APRON CARRIER

the load at the top of its rise as it passes around the head wheels. In the carrier shown by Fig. 16 the pans or buckets, instead of being rigidly attached between the chains, are suspended from them by trunnions, thus always maintaining their horizontal position and permitting them to carry their load successively, horizontally, vertically, horizontally again, or in any given path till the buckets encounter

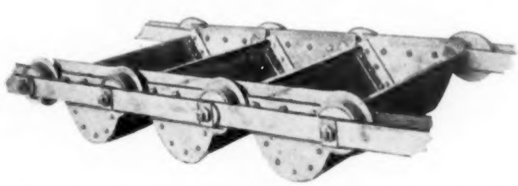


FIG. 14 FINAL DEVELOPMENT FROM APRON TYPE

suitable mechanism for tilting and discharging them at as many points as may be desired. The further step in the perfection of this type of carrier was the addition of lips to the buckets which overlap so that they may be filled without spilling between them. This lap is preferably made Λ-shaped to avoid the carrying of any material and its spilling at the corners where the path of travel changes, and the buckets separate.

23 The overlapping feature makes it necessary, in order that this carrier may complete its path without the effect of the action of gravity being interfered with by the locking of the lips, to provide some device external to the carrier itself to reverse the lap before it starts to ascend, unless as in the most recent form of carrier, correct action is attained by suspending the buckets from extensions of the inner chain links beyond the points of articulations, as in Fig. 16. By this means the buckets travel through a longer path than do the chains from which they are suspended, when passing around the wheel in starting to ascend, and thus automatically unlap themselves. Conveyors of this character are usually run at speeds of 30 ft. to 50 ft. per min., and where they are equipped with self-oiling rollers of 6 in. to 8 in. diameter require but very little power for their operation,

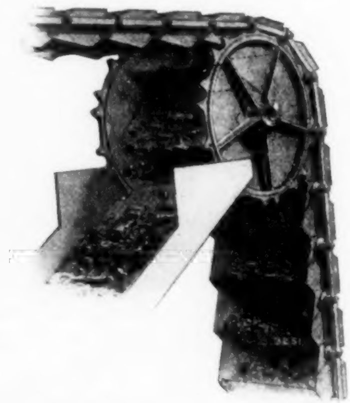


FIG. 15 ROTARY DISCHARGE HEAD

aside from the theoretical power actually required to elevate the material. Table 8 shows the capacity and power consumption of five carriers of this type.

24 Under the head of conveyors which carry their load come those which lift vertically and are known as elevators. The most elementary form consists of one or two endless strands of link-belt, rubber or canvas belt, to which are attached at intervals buckets or cups. The lower wheels are carried in a boot into which the material to be elevated is delivered, and in passing around these wheels the buckets scoop up their load throwing it out upon a chute by centrifugal force as they pass around the upper or head wheel. Friction encountered in pulling through the material in the boot and the speed necessary to get the proper centrifugal effect at the top imposes the

same limits of usefulness upon this form of elevator as upon the scraper conveyor.

25 A type of wider adaptation has continuous or actually overlapping buckets, as shown by Fig. 17. Here the material may be delivered directly into the buckets and in passing over the head the front of each forms a chute for the material in the next following bucket, so that but little centrifugal action is required and the speed may be slow. The most modern form of this elevator is shown by Fig. 18, in which the buckets are at all times in their circuit in actual contact, preventing any material from falling between them, either

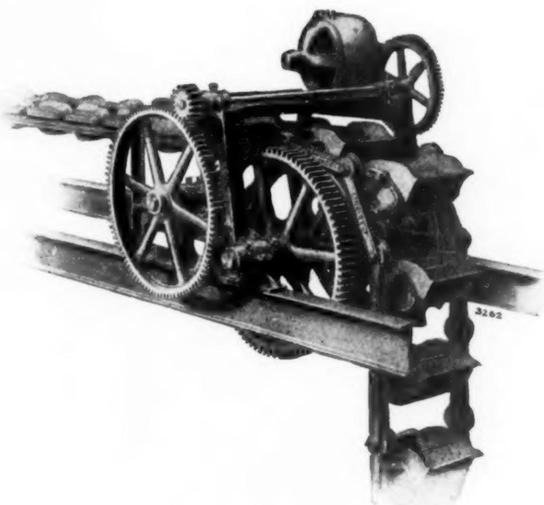


FIG. 16 PIVOTED OVERLAPPING BUCKET CARRIER

at the feeding or delivery points. This is due to their attachment between their chains with the back in the pitch line, so that in passing around the wheels they assume chordal instead of tangential positions,

26 Several of the types of conveyors shown may be intentionally or accidentally loaded beyond their normal working capacity for a short time without serious results. Those which elevate as well as convey, however, are definitely limited in capacity by what the buckets can actually hold without spilling; so that some means of regulating the supply is essential to well designed machines of this class. There are various ways of doing this. Some materials, like

the smaller sizes of anthracite coal, grain, sand, etc., will maintain an even flow through a gate opening which serves to regulate the amount.

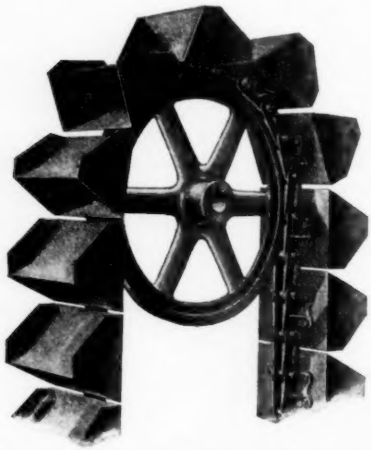


FIG. 17 CONTINUOUS BUCKET ELEVATOR

FIG. 18 OVERLAPPING BUCKETS

For the smaller sizes of soft coal, broken stone, and kindred materials, a rotary feeder resembling a paddle wheel, usually with three or four compartments, is used, as in Fig. 19

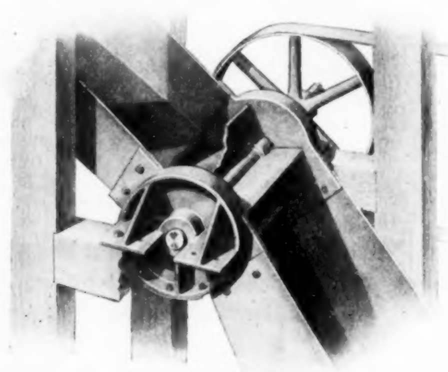


FIG. 19 ROTARY FEEDER

27 For the more coarsely broken materials, such as the larger sizes of coal, and particularly mine-run coal, the supply is regulated

by a feeding belt usually of the type shown by Fig. 12, already referred to, or by the reciprocating plate illustrated in Fig. 20. Both of these devices work beneath a hopper in which is provided an opening of sufficient size to permit the passage of the largest pieces without danger of blocking the opening, the size of this being entirely irrespective of the *quantity* of material it is desired to feed. Both types of feeders must extend beyond the natural angle of flow of the material through the opening, which thus can have no further movement, except as imparted to it by the forward movement of the feeding conveyor, or by the reciprocating movement of the plate.

MERCHANDISE CARRIERS

28 The foregoing has dealt mainly with materials handled in bulk. Many of the same devices with little or no modification will handle materials in boxes, bundles, barrels or sacks. To avoid injury to the packages, they are usually carried on moving wooden aprons attached between two chains, either sliding on ways or with rollers

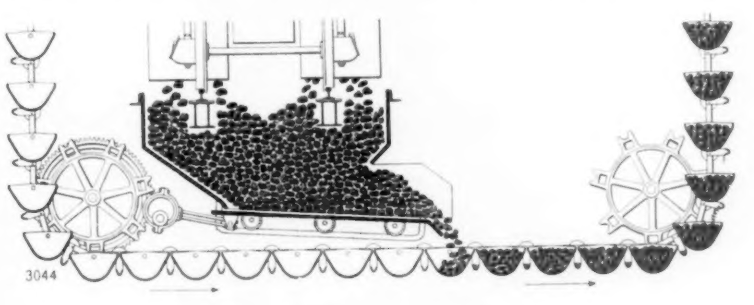


FIG. 20 RECIPROCATING FEEDER

attached if the carriers are heavy ones. Belt conveyors are also very successfully used for light loads and boxes may be conveyed long distances by running over a succession of small rollers set in frames inclined about $\frac{3}{4}$ in. per foot, gravity supplying the conveying force.

29 Packages and barrels are elevated by fingers, arms, or cradles projecting from the chains, arranged to deliver over the head, or of special design that may be tripped and deposit their load at intermediate floors. The most useful form of elevator, however, has suspended trays pivoted between two chains and free to maintain,

their horizontal position under the influence of gravity. These trays have carrying fingers which intermesh with corresponding fingers interposed in the path of the tray at the loading points, and from which they automatically pick up their load, depositing it automatically on similar intermeshing fingers interposed in their descending path at the floor where delivery is required. This form of elevator is shown by Fig. 21; it may follow any prescribed path, horizontally or vertically.

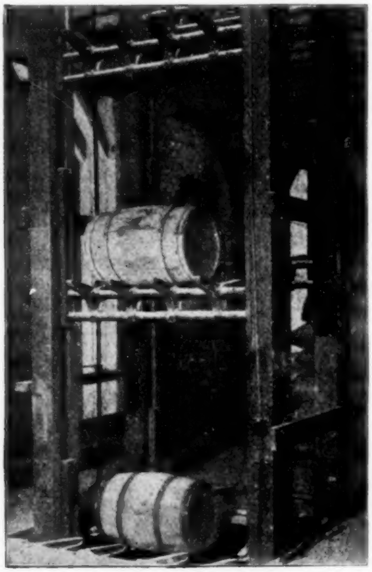


FIG. 21 FREIGHT AND PACKAGE ELEVATOR

CHAINS

30 Inasmuch as chains are an important element in most of the devices under discussion, some brief reference should be made to them. Every common type has been used for conveying purposes, and great ingenuity has been displayed in developing special forms. The present tendency, as might be expected, is towards the use of fewer types, which have demonstrated their fitness for their particular duty. For the conveyors and elevators of lighter service, the Ewart Link-Belt still prevails, or its equivalent in malleable chains with closed ends and pin connection. For double strand service, where the chains are supported, they are made with rollers, as in Fig. 22,

which shows an approved form. The rollers also serve to reduce wear on sprocket wheel tooth.

31 Fig. 23 is a type of chain, which, running in a trough, acts as a conveyor without any attachments, a modification of this having short wings cast integral with and projecting from the side bars, so as to increase the width and conveying capacity. These chains are buried

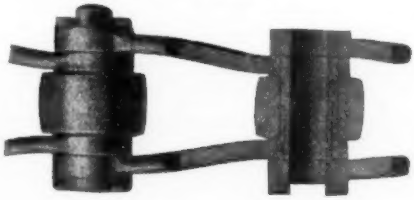


Fig. 22 Roller Chain

in the material conveyed and move it along bodily, up to a depth where the friction on the trough sides would become so great that the chain would pull through the mass instead of moving it.

32 Fig. 24 is a chain long used in two, three, four or more parallel strands for handling logs, bales, barrels, etc., the chains merely sliding, and themselves forming the complete conveyor.

33 For long runs and heavy duty, cable chains have been used, on account of their strength. It is not easy, however, to attach any-

Fig. 23 Sawdust Chain

thing to them, nor does the ordinary pocket or rag wheel prove satisfactory or durable for driving. To meet both of these objections, the Dodge chain, Fig. 25, was devised, the bearing block interposed between the links at their articulation affording ample driving contact for sprocket tooth, and when cast with various attachment wings, providing convenient fastening for flights, etc. This chain also has

the advantage of permitting turns in both horizontal and vertical plane in the same conveyor.

34 To a limited extent effort has been made, particularly in long conveyors, to take advantage of the great strength with relative lightness of the steel cable, as a means for conveying, small cast blocks being clamped to it at intervals to engage the sprocket teeth and carry the flights. The lack of durability of the small individual outer strands of the cable, and the heavy tension under which it has to be run to avoid kinking, are disadvantages that have made its use quite limited.

FIG. 24 TRANSFER CHAIN, DETACHABLE

35 The "Monobar" chain, Fig. 26, has some of the advantages and avoids some of the objections to the cable. It is made up of round steel bars of high tensile strength, with malleable iron connections or sockets, providing a large articulating surface.

36 Fig. 27 and 28 illustrate a successful type of chain designed to handle such gritty materials as ashes, coke, cement, clinker, etc. Here the articulation takes place between a pin and bushing, both case-hardened and both renewable. The bushing on the outside is encased by a sleeve which protects it from contact with the sprocket

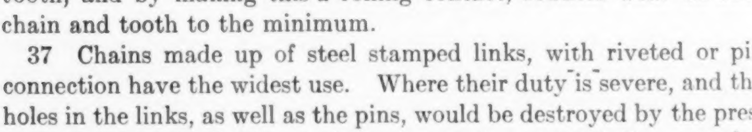


FIG. 25 DODGE CHAIN

tooth, and by making this a rolling contact, reduces wear on both chain and tooth to the minimum.

37 Chains made up of steel stamped links, with riveted or pin connection have the widest use. Where their duty is severe, and the holes in the links, as well as the pins, would be destroyed by the pressure on the articulating surfaces in passing around the sprocket wheels, the inner pair of links is connected together by a rigid bushing and the pin held rigidly in one or both of the outer pair of links.

All relative movement, therefore, must take place between the pin and the bushing, and a large bearing surface is thus obtained, without any wear coming upon the links themselves. Fig. 29 shows a typical chain joint of this kind.

38 The Maximum chain, Fig. 30, represents, perhaps, the highest development of a chain designed merely for conveying purposes.

Fig. 26 Monobar Chain

Here it will be seen that both outer and inner links have a bearing upon the pin for the full width of the chain, and it being unnecessary to secure the pin to either set of links, it is free to roll to a greater or less extent each time the chain turns around the wheels. This seems to have a densifying rather than wearing effect on the pin, and always keeps it perfectly round. In chains of this type it is not unusual to find the pins highly polished after considerable use.

APPLICATIONS OF CONVEYORS

39 As bearing on the uses of conveyors and the economy that may be effected, several different applications of conveyors will be discussed and certain installations referred to.

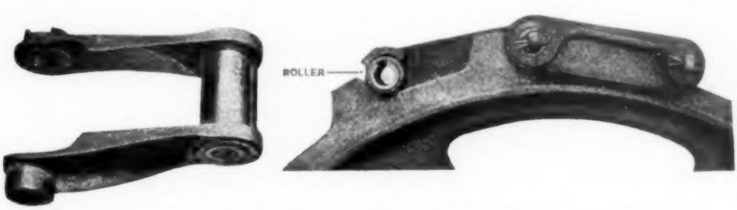


Fig. 27 and 28 Chain for Gritty Materials

40 The simplest application is merely to save labor expense in transporting material between given points. A conveyor, Fig. 31, handling 700 lb. glucose barrels from refinery to wharf 450 ft. distant, actually effects a very great economy—the cost of this installation, some \$3000, being saved in approximately three months. As this

was installed some years ago the first cost would be much greater today, but the proportionate increase in the cost of labor would make its showing equally good.

41 Short conveyors similar to the above are used to handle merchandise between vessels and warehouse, being hinged at the warehouse floor level, so that they may be inclined upward or downward, according to the position of ports of the vessel, at any stage. The purpose of such an installation is to aid labor rather than displace it. Nevertheless these conveyors are incidentally money-savers. Four of them operating in pairs running in alternate directions, each 110 ft. long, saved their first cost, approximating $16 000 in one year of use.

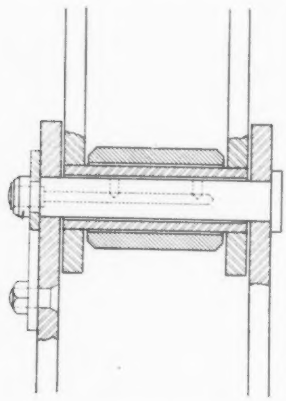


Fig. 29 Special Chain Joint

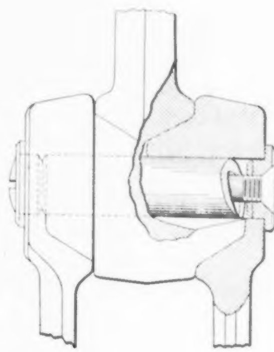


Fig. 30 Maximum Chain

42 Elevating and conveying devices have an extended use as an adjunct to special processes where the mere conveying function is secondary. Illustrations of this are conveyors which, at slow speed, carry ores through roasting furnaces, plate glass and glass ware through annealing ovens, and various products through drying, steaming, or sterilizing chambers and tanks. Materials are also conveyed for the purpose of giving them time to cool, and slow moving apron or platform carriers are utilized for the sorting and picking of ore and coal, and for filling, sealing, and labeling canned and bottled goods.

43 The broader and more intelligent use of conveying machinery which brings it directly within the province of the engineer, is where in addition to its primary purpose, it determines or modifies the character of a plant. The accepted type of modern power house with large

storage bins above the boilers, from which the coal is drawn by spouts to mechanical stokers, is made possible by the conveyor which receives the coal from cars, boats, or wagons, outside the building, elevates and distributes it in the overhead bin; and into the returning run of which in the basement, when not handling coal, the ashes may be drawn from hoppers beneath the boilers and elevated to overhead pockets, whence they may be drawn as convenient. In such boiler rooms no coal or ashes are visible, and the same cleanliness may be maintained as in the engine room.

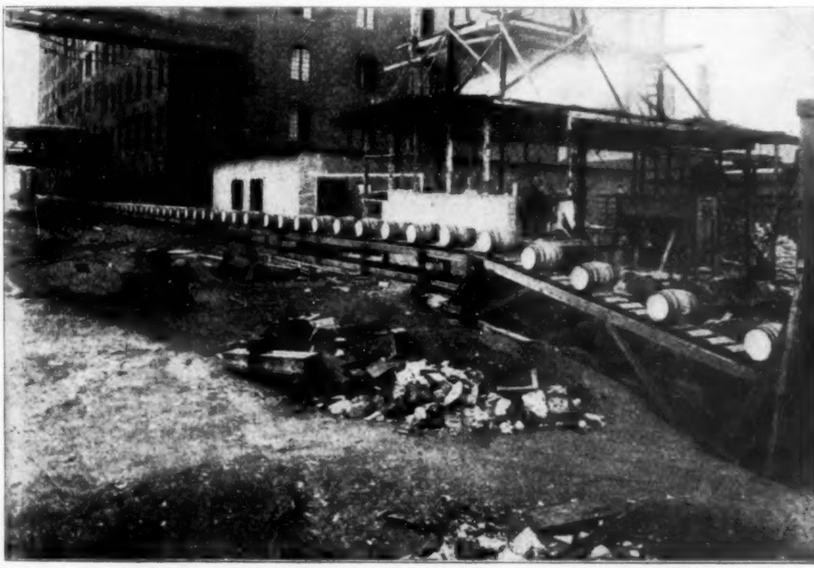


Fig. 31 Freight Carrier

44 The pivoted bucket carrier, to which attention has been called, has for a number of years been practically regarded as standard for this purpose, in the majority of cases handling both coal and ashes, but in some large power houses coal only, the ashes being taken care of by a second carrier, or by some other means. While ashes are a much more destructive material to handle than coal, largely due to their corrosive effect when wet, the cost of maintenance of these carriers is so small, even under this condition, that they are well adapted to the double service.

45 Taking as an illustration a light power plant of moderate size in New Hampshire where one of these carriers had been in use four years with no serious interruption: the annual consumption was 18-000 tons of coal per year, elevated 52 ft. and carried horizontally 110 ft. The average operation was 60 hours per week, one man looking after the carrier in connection with his other duties. The cost of repairs averaged $120 per year, or 6/100 cent per ton handled, with

FIG. 32 ELEVATING COAL TO TOP OF BREAKER AT MINE

a cost for power and attendance of 3 cents per ton, making a total of 3-6/100 cents per ton. These unit costs are high because the coal is handled at a rate very much below the capacity of the machine, and no credit in these figures is given to the tonnage of ashes handled.

46 The street railway power house referred to in Table 8 was equipped with duplicate carriers and consumed 15 000 tons of coal per month, the carriers operating about 65 hours per week, with a cost for repairs of but 1-10 cents per ton; and the machines showed

scarcely any wear. The cost per ton for power, supplies and attendance was 1-1/10 cents per ton.

47 A carrier equipped with 24 by 24 buckets at the Union Stock Yards, Chicago, with a vertical lift of 58 ft. and a horizontal run of 138 ft. has been in continuous operation since 1903 handling an average of 2500 tons of coal per week with no expenditure whatever for repairs and is still in such excellent condition that it is probable no repairs will be required for some four or five years.

48 The modern coal mine affords another illustration of plant modification. Here at a little distance from the shaft mouth is a building of great height necessary for the coal to pass through the various preparing operations of crushing, screening, picking and sorting, into the shipping pockets. Formerly the mine cars had to be elevated above the shaft to the height of the building, pushed from the cages, and hauled to the dump at the top of the building, and back to the cages. The modern practice is to dump the cars without removal from the cages, at the shaft mouth, into an inclined conveyor of great capacity which carries the coal to the top of the breaker or tipple building, permitting the immediate return of the mine cars, when dumped, to the bottom of the shaft. An illustration of such a conveyor is shown in Fig. 32. Such conveyors handle coal at the rate of 500 tons and more per hour, and similar ones used in loading vessels have successfully and frequently handled 1000 tons per hour.

49 One of the earliest of these carriers was placed in operation late in 1902 near Pittston, Pa., operating on an incline of 25 deg. and conveying the coal 355 ft. the buckets being 48 in. wide. This carrier handling 130 000 tons per month was in excellent condition after four years' use, the only repairs being the renewal of the carrying rollers, and the driving pinions of the head gearing. The cost of repairs averaged for material 4/100 cent per ton handled, and for labor 6/100 cent per ton, or a total repair cost of 1/10 cent per ton. This machine had a further record of never having caused a day's shutdown of the plant.

50 In the most advanced development and utilization of conveying machinery at the present day, it forms the basis of a complete process or system, two or three illustrations of which will be briefly referred to.

51 In a typical plant for supplying locomotives with coal, sand, and water, and for the removal and disposition of the cinders, the coal is carried in an overhead pocket of 1000 tons capacity, or from one to two days' supply. From this pocket it is drawn as required into

small scale pockets above a series of coaling tracks, so that accurate record may be kept of the amount delivered to each locomotive. While standing beneath the pocket, or adjacent to it, the cinders are raked into small trolley cars in the raking pits which are long enough to accommodate three locomotives each. These trolley cars as filled are pushed over hoppers beneath the station, from which the cinders may be drawn into a carrier running beneath them all, and finally ascending and delivering into overhead cinder storage pocket.

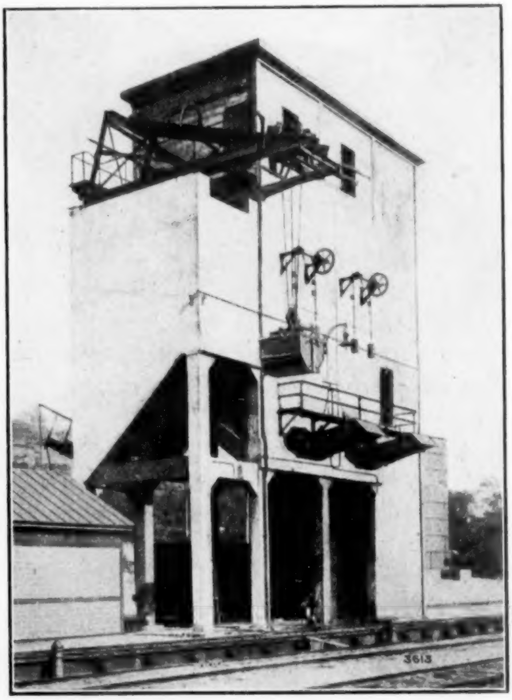


FIG. 33 LOCOMOTIVE COAL AND CINDER STATION OF REINFORCED CONCRETE

Elevators of suitable design receive the coal and sand from supply tracks and deliver into conveyors which distribute into the overhead coal and sand pockets. As a safeguard against accident and delay, the coal receiving track hoppers and elevators are in duplicate.

52 In Fig. 33 is a locomotive coal and cinder station of reinforced concrete, built by the Link Belt Engineering Co., for the Pittsburg and Lake Erie Ry., Pollock, Pa. The capacity of the coal pocket

is 200 tons, which is supplied by a gravity discharge elevator. The latter is fed from track hopper, as shown in diagram, and delivery to boot is made regularly and automatically by reciprocating feeder. The cinder bin holds 20 tons and is served by a tub hoist of 1-ton capacity. The tub rests on a track, running in a pit under the service track and receives cinders from the fire-box while the locomotive is coaling.

53 Cost figures of fueling stations are somewhat unsatisfactory for purposes of comparison as the practice of railroads in what they charge against cost differs so greatly. Table 9 shows the cost per ton of coal handled at twelve stations located on one western road for a period of six months, these figures covering labor cost only.

TABLE 9 COST OF HANDLING COAL AT 12 FUELING STATIONS

Station	February	March	April	May	June	July	TONS HANDLED		
							May	June	July
1	0.102	0.108	0.027	0.036	0.014	0.019	2836	3026	2807
2	0.025	0.027	0.033	0.031	0.033	0.033	1137	1058	1072
3	0.012	0.011	0.023	0.027	0.031	0.012	9063	9028	8807
4	0.017	0.016	0.020	0.029	0.029	0.026	1222	1145	971
5	0.009	0.009	0.010	0.015	0.013	0.010	4156	4308	4044
6	0.007	0.006	0.011	0.012	0.013	0.013	2330	2479	2333
7	0.011	0.009	0.010	0.008	0.009	0.011	2939	2680	2156
8	0.011	0.010	0.012	0.025	0.013	0.013	2157	1976	1845
9	0.014	0.014	0.012	0.012	0.012	0.018	2360	2448	2501
10	0.038	0.011	0.011	0.018	0.009	0.010	3808	3875	4488
11	0.017	0.014	0.014	9.017	0.020	0.041	1595	1403	1359
12	0.033	0.020	0.021	0.020	0.030	0.010	1795	1909	2355

Average cost of handling the coal at twelve stations for
A period of six months $1\frac{3}{4}$ cent per ton.

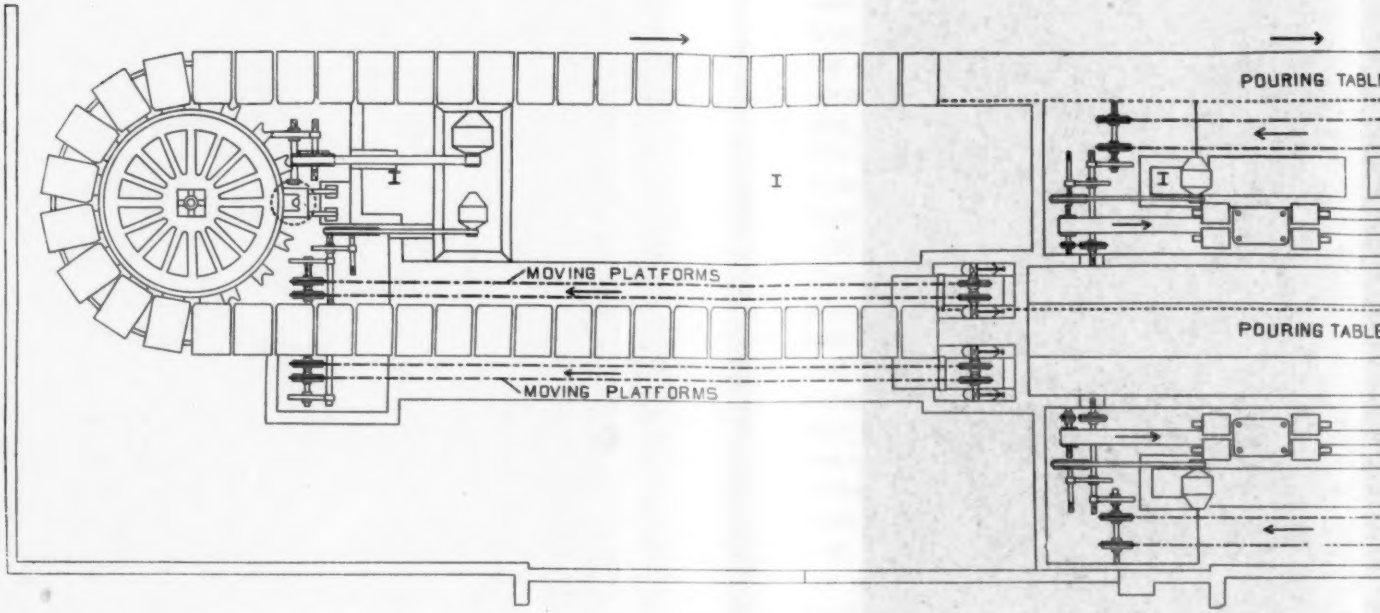
The variation is due to the tonnage handled and to the fact that while for the most part the coal is received in 50-ton self-cleaning cars, occasionally it had to be *shoveled* out of flat bottom gondola cars. Where the cost exceeds 2 cents per ton it is due to this cause. The total average shows a cost of $1\frac{3}{4}$ cents per ton.

54 Table 10 shows comparative records for one year of eight locomotive coaling stations and coaling trestles of a large eastern railroad. The average total expense for the pockets equipped with steel elevating machinery is 4.88 cents per ton; for the inclined belt conveyor type it is 9.69 cents per ton, and for the trestles 4.46 cents per ton. However, the last figure does not take into account any allow-

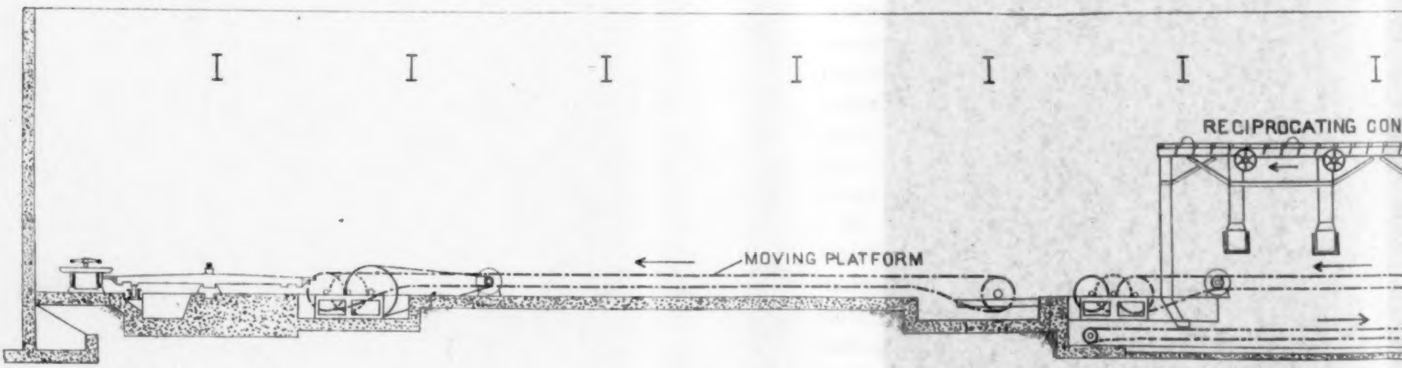
TABLE 10 OPERATING AND MAINTENANCE COSTS FOR LOCOMOTIVE COALING STATIONS FOR ONE YEAR

Type	Tons handled	Repairs labor and material		Switching charges		Labor dumping coaling		Fuel for operating machinery		Total expense	
		Total	Per ton	Total	Per ton	Total	Per ton	Total	Per ton	Total	Per ton
Bucket elevator	130 850	$275.48	$0.0021	$924.00	$0.0071	$4394.44	$0.0336	$826.00	$0.0063	$6419.92	$0.0491
Inc. belt conveyor	62 899	1485.97	0.0236	440.29	0.0070	3652.61	0.0581	494.00	0.0079	6072.87	0.0966
Bucket elevator	43 321	320.85	0.0074	456.25	0.0105	996.45	0.0230	378.77	0.0087	2152.32	0.0496
Bucket elevator	183 410	2415.22	0.0132	1265.53	0.0069	4536.77	0.0241	550.00	0.0030	8767.52	0.0478
Inc. belt conveyor	47 109	1456.71	0.0309	334.47	0.0071	2473.70	0.0525	316.00	0.0067	4580.88	0.0972
Trestle	110 138			813.60	0.0074	4631.78	0.0421			5445.38	0.0495
Trestle	276 397			2034.60	0.0074	8451.56	0.0305			10486.16	0.0379
Trestle	61 570			800.41	0.0130	2056.85	0.0334			2857.26	0.0464

Average total expense per ton handled—stations with inclined belt conveyors $0.0969
Average total expense per ton handled—stations with elevators 0.0488
Average total expense per ton handled—stations with coal trestles (exclusive of repairs) 0.0446

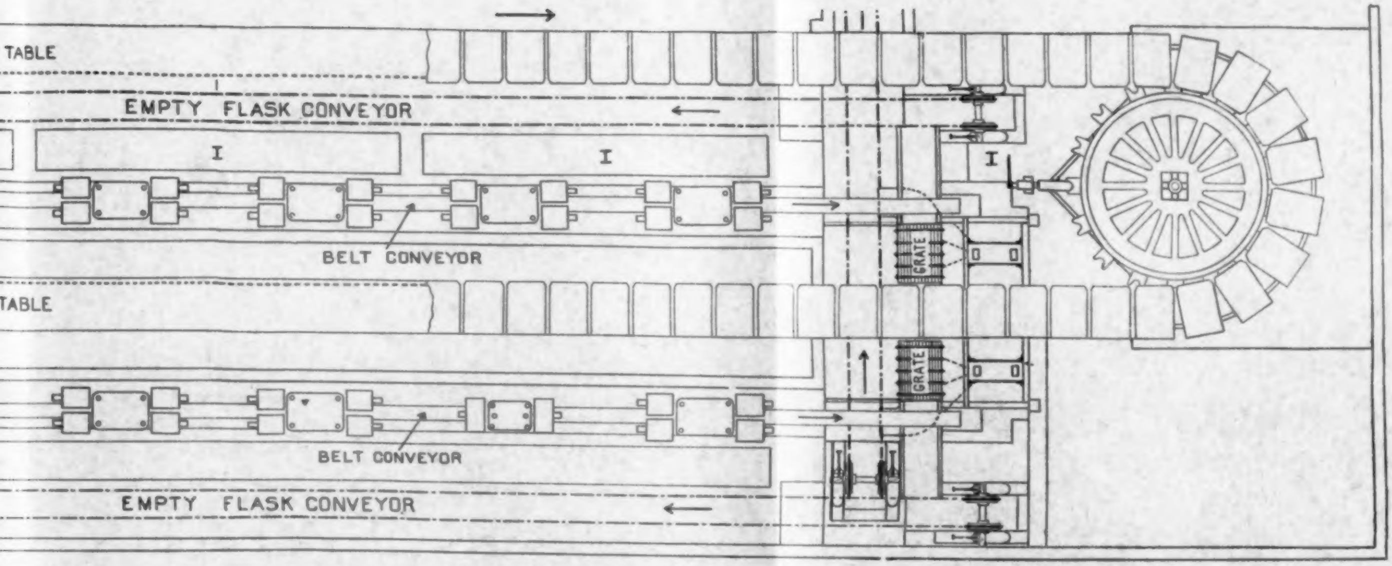


Plan



Elevat

FIG. 34 CONVEYOR SYSTEM



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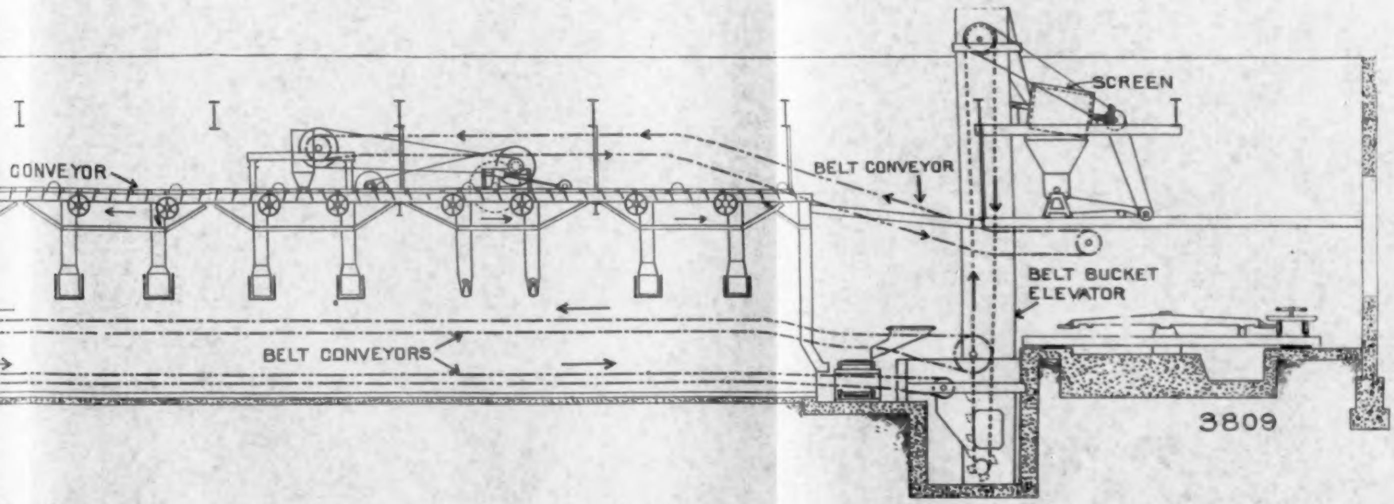


Elevation

STEM FOR MODERN FOUNDRY

ance for maintenance of the trestle. This charge will be small for the first 12 or 15 years, after which it will be considerable. The switching charges are about the same for the trestle and mechanically operated pocket. The chances for accidents to locomotives and cars are considerably greater in drilling cars on and off the trestle. The trestle, of course, has no expense for fuel. The chief difference in costs between the inclined belt conveyor stations and those with steel conveying machinery lies in the repair charges.

55 An important application of conveying machinery is the well-known Dodge system for the storage of anthracite coal. Conical piles of coal are formed by conveyors supported on shear trusses. This, now the recognized standard method for the storage of anthracite coal in large quantities, has reduced the handling costs in and out of storage, which formerly were 20 cents per ton and upwards to less than 4 cents per ton, and by making the daily output of the mines to a considerable extent independent of the demand of the moment, worked a benefit both to the operators and to labor by establishing more uniform working hours.

56 Fig. 34 shows a complete handling system of a modern foundry making a uniform product in large quantities. A conveyor, which is practically a continuous moving table, runs the entire length of the foundry. Near one end of this stand the molding machines, and a mold when made is immediately placed on the carrier, which is constantly moving at a slow speed. Near the other end of this carrier stand the cupolas, and as the molds arrive there, they are poured by men who perform no other function, the work of pouring being facilitated by the suspending of the ladles from overhead trolley rails, and by conveyors alongside the main carrier and flush with the floor, moving at the same speed as the carrier, upon which the pourers stand while pouring.

57 The returning side of the carrier is filled with the molds which have been poured, and which have time to cool. Before the molding machines are reached, the molds are automatically raked from the carrier onto screen bars, through which the sand falls. The castings slide over these bars and fall on a conveyor which takes them to the cleaning room; the flasks are lifted off and placed upon other conveyors, which return them to the molding machines. The sand, which has fallen through the grating, is caught by elevators, delivered to screens which remove lumps and sprue, passed through mixers onto cooling and tempering conveyors and is finally delivered to conveyors running above the molding machines which furnish the sand in

requisite quantities to them. So completely is the handling of all the products carried out without manual labor that the little sand "struck off" the molds before putting the two halves together is caught by small belt conveyors beneath the floor and carried to the sand elevators.

58 One of the earliest foundries to adopt this system of handling, although already having a systematic organization, and taking every advantage of highly specialized molding machinery and piecework rates, was able to effect a saving of ½ cent per pound; and very careful and conservative estimates made on the probable economy of a similar system by a foundry of large and rather a complex output, contemplating such an installation, showed that its first cost, of about $45 000, would probably be saved in 2½ years.

59 These last few illustrations of the most systematic use of conveying machinery, bring it peculiarly within the province of the engineer. In these no expense has been spared that could add to the efficiency, reliability and durability of the installations, and the economies effected thereby have been correspondingly great. While such results may not be hoped for from every installation, these may at least suggest to engineers that the subject is one that will well repay their study.

NOTE—The discussion on this paper and the Author's closure are published under "Discussion on the Conveying of Materials," No. 1195 in this volume.

—EDITOR.

No. 1192

THE BELT CONVEYOR

By C. Kemble Baldwin, Chicago

Member of the Society

EARLY HISTORY

The Transactions of this Society contain but one paper treating of the handling of material by belt conveyors, this one being contributed in 1885 by Mr. T. W. Hugo upon Belts as Grain Conveyors. Mr. Hugo closes his paper as follows: "Enough has been said to show that it is a satisfactory method of handling grain, and not grain alone, but there are various articles that could be expeditiously and cheaply handled by this plan, or with modifications of it, to suit the articles to be handled."

2 Although little has been written on the subject, the prediction made 23 years ago has been realized, and today millions of tons of ore, stone, earth, coal, etc., are being successfully moved on belt conveyors; and this type of conveyor has firmly established itself in a field of its own.

3 In 1890 Thomas Robins, Jr., of New York, then engaged in the manufacture of rubber goods, visited the Edison Mines in New Jersey, where he saw rubber belts used for handling iron ore in the concentrating plant; and while they handled enormous quantities of ore with very little power, the belts, the best they could buy, lasted but a few months.

4 Mr. Robins subsequently conducted a long series of experiments to determine the proper belt construction, and the best compounds of rubber to withstand the abrasion of the material carried. Small rubber samples, each mixed differently, were exposed to the action of a powerful sand blast for the same length of time, each sample being carefully weighed before and after the test in order to note the loss in weight. Test samples were also nailed to boards and placed in

Presented at the Detroit Meeting (June, 1908) of The American Society of Mechanical Engineers.

streams of falling ore in such position that the ore was immediately deflected. These samples were also weighed before and after each test.

5 These tests, which extended over a period of nearly two years, showed that certain adulterants could be used in sufficient quantities to bring the cost of the belts to a marketable figure and still retain their abrasion resisting qualities. Belts made in 1892 from these compounds wore for over four years under exactly the same conditions which had destroyed the former belts in three months' time. In 1896 Mr. Robins read a paper before the American Institute of Mining Engineers in which he fully explained the belt conveyor of that day and outlined his experiments.

6 The oldest type of conveyor consisted of a belt dragging along in a wooden trough and passing over flat faced end pulleys as in Fig. 1. The belts were of single ply canvas, or were of the ordinary type

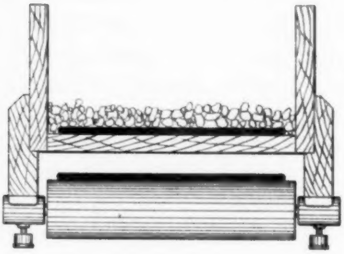


Fig. 1

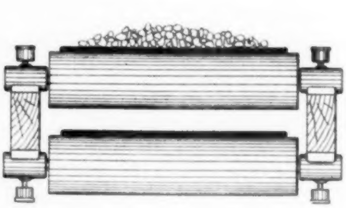


Fig. 2

Flat Faced Rollers for Belt Conveyor

of rubber or stitched canvas belts. These crude home-made affairs, while they did the work, wore out quickly, which made the cost of belt renewal very high.

7 Flat wooden rollers, Fig. 2, were then substituted for the trough and the belts made wider to keep the material from jarring off when passing over the rollers. In many cases skirt boards were placed the entire length of the conveyor to keep the material from rolling off, as in Fig. 3.

8 Naturally the belt would sag between the rollers, resulting in the material catching between the belt and skirt board at the rollers, which soon cut strips off the edges of the belt, and the skirt boards were arranged so they could be moved in as the belt became narrower.

9 The next and most natural development was to turn up the edges of the belt, Fig. 4, forming a trough which not only kept the

material from rolling off, but prevented the belt from sagging between the supporting rollers. The wooden spool was made either in one or three pieces. Fig. 5 shows the concave roller made variously, in one piece, or cut into sections as indicated below, in order to reduce the slip on the under cover of the belt.

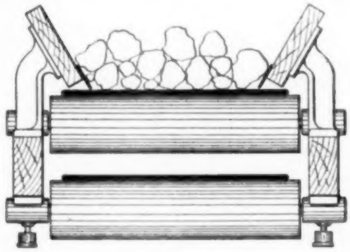


Fig. 3 Belt Conveyor with Skirt Boards

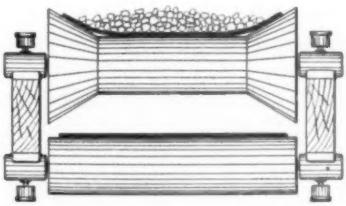


Fig. 4 Wooden Spool to Trough the Belt

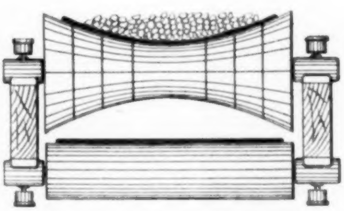


Fig. 5 Concave Roller

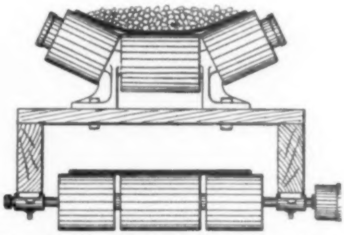


Fig. 6 Idler for Belts not Over 26 in. Wide

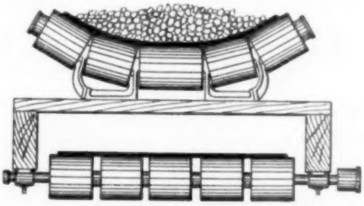


Fig. 7 Idler for Wide Belts

10 In the case of both the spool and the concave roller this slip, due to the different diameters, while not serious on a grain conveyor where the belt presses lightly against the rollers, is a serious matter

when handling the heavier materials, as the slip soon destroys the under cover of the belt.

11 A belt running at 400 ft. per minute 8 hours per day and 300 days per year, will travel during the year 57 600 000 ft. Taking one section of the roller in Fig. 5 with small diameter of 8 in. and large diameter of 12 in., assume that the section will turn at the speed of the mean diameter of 10 in. It will, during the year, make about 22 000-000 revolutions and with a slip of 6 in. per revolution, there will be a total slip of about 2000 miles per year. Results of actual practice

Fig. 8 36-in. Conveyor Before Belt was Put On

prove how destructive this is and stress is laid on the faults of this form of construction because several patents have been issued recently on carrier pulleys of this general type.

12 All of the forms described above are still used to a very limited extent, but the idler used almost universally today consists of a series of small pulleys of equal diameter arranged in the same or different planes, as in Fig. 6 and 7, which will produce no wear due to slip.

13 The troughing idlers should be spaced as follows, depending

upon the weight of the material carried: On 12-in. to 16-in. belts, 4½ ft. to 5 ft. apart; 18-in. to 22-in., 4 ft. to 4½ ft.; 24 in. to 30 in., 3½ ft. to 4 ft.; 32 in. to 36 in., 3 ft. to 3½ ft.

14 Fig. 8 is a view of a 36 in. conveyor 670 ft. from center to center before the belt was put on. This conveyor has been in service over three years and has a capacity of 500 to 600 tons per hour.

15 The lubrication of the idlers is an important matter, particularly where rubber belts are used, as oil or grease soon causes the rubber to deteriorate. Oil should not be used as it leaks out when the conveyor is not running and soon coats the faces of the pulleys and the belt. As conveyors usually work in dirty places and it is impossible to keep the oil wells clean, grease lubrication through hollow shafts is generally used. Such bearings are practically dust proof, any foreign matter being carried out by the grease and the small ring of grease which forms on the hubs of the pulleys acts as a most efficient dust collar. Grease lubrication without question increases the power consumed, but the great advantage of clean bearings, coupled with the low power requirements of the belt conveyor, has resulted in the general adoption of this method of lubrication.

16 The return belt is always carried on straight idlers consisting either of a number of small pulleys turning on a hollow shaft with grease cup on one end, or of small pulleys keyed to a solid shaft turning in pillow blocks. They should be spaced from 8 to 12 ft. apart, according to the weight of the belt.

BELTS

17 Belts made of a great variety of substances have been used on conveyors and many tales of wonder have been told of the performances of some of them. A careful research, however, shows little that is really new or important to have been added to the art, although vast improvements have been made in the older types.

18 Stitched canvas and woven cotton belts are used to some extent, but generally their service is not satisfactory, mainly because it is impossible to water proof them thoroughly. Belts of this class are usually filled with oil, paraffin, or other such substance, the outside being coated with some water proof paint. When new, they are fairly flexible, but soon become so stiff that they will not trough properly, and it is difficult to make them run true and straight. They are sensitive to atmospheric changes, requiring constant adjustment of the take-ups to keep them at proper driving tension. In one

case under observation a 24-in., 5-ply canvas belt on a conveyor 400 ft. centers was handling crushed stone when the heat of the sun caused the belt to stretch and slip on the drive pulley. The take-ups were adjusted and the conveyor was again started. During the noon hour there was a shower of rain, causing the belt to shrink and as it was already tight, the tail structure of the conveyor was pulled down by the contraction. The belt was guaranteed water proof, or as the manufacturers said, "Commercially water proof."

19 The main claim of these belts is low first cost, but they should not be used on permanent installations, particularly where out of doors, or where subject to atmospheric changes.

20 The rubber covered belt is more generally used than any other type, and the belt conveying industry has been built up mainly on the remarkable showing of the rubber belts made by Mr. Robins, based on the tests described previously. While belt conveyors had been used for many years for handling grain, and to some extent for other materials, the apparatus was crude and mostly home made, so that the real development may be said to have started with the perfection of a durable rubber belt.

21 Briefly, rubber belts are made as follows: The duck is run through the calender which consists of a series of heated rolls. This machine coats the duck with rubber, known as "friction." This frictioned duck is then cut into the proper widths and the belt is laid up to the desired number of plies, the operator using a hand or power roller to cause the plies to stick together. The cover is then put on and the belt put in the vulcanizing press where it is subjected to heat and pressure. The belts are stretched before being vulcanized.

22 It will be noted that the belt consists of three parts: First, the duck which gives tensile strength; second, "friction," which cements the plies of duck together; third, the rubber cover which protects the duck from moisture and abrasion.

23 Cotton duck used in the manufacture of belts is made in bolts about 375 to 400 ft. long. Belts longer than this should be joined with metal lacing. Internal splices should be avoided, and special lengths of duck, while sometimes used, generally cause delay and increased cost and are not to be recommended, especially in wide belts, on account of the increased weight of the roll of belting to be handled in shipment and erection.

24 The average manufacturer of rubber goods will, in making up belts, cut his duck so as to have the least waste, as the narrow strips are almost valueless. This results in the longitudinal seams in the

belt being located at random. A careful study of hundreds of belts operating under a great variety of conditions has shown the great importance of the proper location of these seams. When properly placed, they cause no trouble, while many belts made by reputable rubber companies have gone to pieces in a few months because the manufacturer did not understand the conditions under which the belts must operate, and naturally built them only with regard to his factory economy as a guide to construction. The real life of the rubber belt is in the cover, provided of course that the belt is properly laid up, the friction is good, and the cover adheres properly.

25 A thin layer of rubber forming the cover of a belt under observation handling iron ore was found to represent one-half the life of the belt, although forming less than one-fifth the total thickness. The function of the cover is to protect the body of the belt from moisture and abrasion. It must therefore be soft, pure and durable, and still not be too high in cost. The compounds forming the covers were determined by the sand blast and ore tests previously described and since the time of these tests the compounds have been greatly improved by a careful study of belts in actual service.

26 Much harm has been done to this industry by the fact that the buyer has not, as a rule, a very extensive knowledge of rubber and he cannot, from the samples submitted by a rubber company, be sure that he is getting a belt suited to the work to be performed. The specialist in the manufacture of belt conveyors has his business at stake when he offers his product, and should the belts fail his whole business suffers. The ordinary salesman of rubber belts on the other hand does not, as a rule, have an opportunity to study the operating conditions of belt conveyors and should the belts fail, he blames the mechanism, while the purchaser condemns the entire system. It is therefore more satisfactory to purchase the complete conveyor from the specialist who knows the conditions and can be held responsible.

27 Rubber belts are absolutely water proof *only when the cover is intact*, as the friction coats only the surface of the duck and is not absorbed by the same. When used in such service that the belts are continually soaked in water, they are seriously damaged by the water reaching the duck through cuts or punctures in the covers. The duck absorbs the water, causing the rubber to lose its hold, thus forming a blister which is quickly extended by passing over the end pulleys. This does not apply so much to belts working out of doors and sub-

ject to wetting from storms, as to those handling dredgings, wet tailings and similar materials.

28 It will be noted that both the cotton and rubber belts lack the property of being *absolutely* water proof. The only belt, which to the writer's knowledge possesses this quality, is the Balata Belt.

29 Balata is a vegetable gum found in Venezuela and the Dutch East Indies. In nature it lies between gutta percha and india rubber, but differs from them in its great tensile strength, freedom from oxidation, and the fact that it does not deteriorate with age. It is dissolved properly in but one solvent, the nature of which is a carefully guarded secret of the manufacturers.

30 The duck used in the Balata belt is woven by a special process on powerful looms, making it a very compact, closely woven, and non-stretching fabric. The Balata is applied to the fabric in a liquid form so that the gum penetrates and saturates every fiber of the fabric, thoroughly waterproofing it. This belt is not only water proof, but it possesses about 20 per cent greater tensile strength than a rubber belt of the same number of plies, due to the more closely woven fabric and the greater strength of Balata as a friction. The fabric is never exposed to the great heat of vulcanization, which somewhat impairs the strength and vitality of the fabric used in rubber belts. For lighter materials, the Balata Belts are used without covers, but when heavy abrasive materials are to be handled a rubber cover is provided to protect the fabric from wear.

31 There are two points which determine the number of plies of duck making up a conveyor belt. First, there must be sufficient tensile strength; second, the belt should have enough stiffness to support the load properly between idlers. With the speed and the power transmitted known, the stress in the belt per inch of width may be determined and this stress should not exceed 20 lb. per in. per ply on the rubber belts, although it may be increased 20 per cent on Balata belts. Where the power required is small, the stiffness of the belt fixes the number of plies. Table 3 gives the maximum and minimum plies for different widths of belt.

32 Belts are sometimes made endless in the factory, but they are difficult to install and nothing is gained, so this practice is not recommended.

33 Lapped splices have been vulcanized in the field with a portable press, but this requires an expert and is very difficult as all moisture must be dried out of the duck, otherwise steam will form when heat is applied, forming a blister in the splice. Lapped cemented

splices are sometimes used, copper tacks being driven through the belt to hold the edges down. All of these methods are necessary in some cases, but generally a metallic lacing like the Bristol, Fig. 9, or the Crescent lacing is the most satisfactory. These lacings are strong, are easily put in and do not seriously damage the ends of the belt,

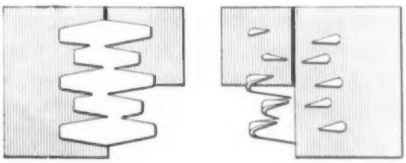


Fig. 9 Metallic Lacing

provided the holes are first punched with an awl. When the belt becomes worn at the edges of the lacing, it is a simple matter to cut the lacing out and put in a new row.

DRIVING MACHINERY

34 The belt conveyor has one great advantage over most other types of conveyors, in that it may be driven from any point in its length. When properly designed, the drive located at the tail or loading end will prove as satisfactory as when at the head. The drive may be located on the return or lower belt at any point between the head and tail ends with satisfactory results. The longest heavy duty belt conveyor ever built (1000 ft. from center to center, handling about 400 tons of material per hour) is driven from the loading end. This feature of being able to apply the power to the conveyor at any point in its length, is of great value in the design of plants, particularly with conveyors of large capacity.

35 Take, for example, the case of a long, inclined conveyor, running up an inclined structure, the drive may be located at the foot of the incline where proper foundations may be obtained and where the heavy parts may be conveniently and cheaply handled. Any other type of conveyor would require a drive located at the top of the incline, greatly increasing the cost and weight of the supporting structure, with corresponding increase in cost of installation. Where one conveyor discharges to another, they may be driven by a single motor or engine at the point of intersection, one being head, the other

tail driven. A conveyor carrying tailings from a mill to a waste pile may be driven from the shafting of the mill by tail drive, thus saving wiring to the head end and an isolated motor.

36 It is frequently desirable to drive one conveyor *through* another, using the conveyor belt to transmit the power. For example, in a system of four conveyors, *A*, *B*, *C*, and *D*, material was fed to *A* which discharged to *B*, *B* to *C* and *C* to *D*. A motor was located at the point where *B* discharged to *C*. Power was transmitted from the head of *B* through the belt, and *A* was connected by suitable gears and chain to the tail of *B*. Similarly conveyor *C* was driven at the tail end and *D* was geared to the head of *C*. It will be noted, therefore, that the four units were geared to a single motor located at a convenient point along the line and arranged so that they all started or stopped at the same time.

37 The drive located at any point in the length of the conveyor may be so built that the conveyor may be reversed, thus carrying material in either direction by simply reversing the motor or engine.

38 The driving machinery for the belt conveyor is extremely simple. Power is applied to one or more of the pulleys over which the conveyor belt passes. These pulleys should have heavy arms and rims with extra high crowns. They should be secured to their shafts by both keys and set screws.

39 Formerly the driving pulleys were increased in diameter as the duty increased, so as to obtain easily a greater arc of contact. In recent years multiple pulley drives have been used on the longer conveyors with most satisfactory results. In this type of drive, the belt passed over two or more pulleys geared together so that they turn at the same speed. This makes it possible to use smaller pulleys, thereby simplifying the speed reduction from the motor or engine. For example, a conveyor with a belt speed of 400 ft. per min. having a driving pulley 48 in. in diameter making 32 r.p.m. and driven by a motor at 800 r.p.m. will require a reduction of 25 to 1. If a multiple pulley drive made of two 24 in. pulleys is used, the reduction will be only 12½ to 1. With a single pulley drive, the belt must always be under the proper driving tension fixed by the take-ups. With the multiple pulley drive, however, the belt may be run as slack as desired provided there is some means of keeping the belt in contact with the second drive pulley, by allowing it to sag or using a weighted take-up. A belt under great tension is hard to train, therefore the conveyor with multiple pulley drive may be more easily adjusted and operated.

40 Pulleys of small diameter should be avoided on the heavy belts, otherwise the constant bending when under heavy stress will cause the friction to lose its hold and destroy the belts. In many cases it is advisable to cover the driving pulley with a rubber lagging to increase the tractive power. This is particularly the case when the conveyor operates in a dusty place.

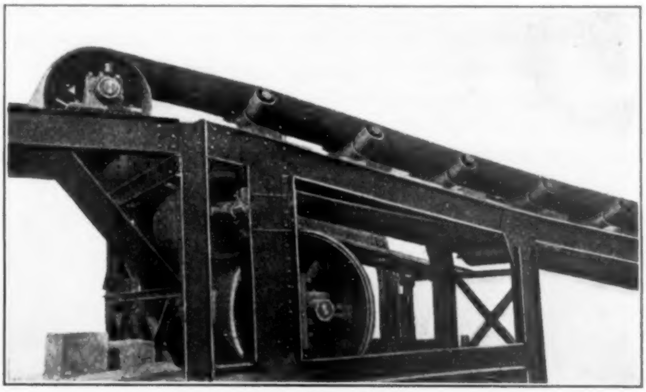


FIG. 10 HEAD DRIVE—DRIVING DONE BY LARGER PULLEY

41 The following table gives the minimum size of driving pulleys which should be used on the various widths of belt dependent on the number of plies:

TABLE I MINIMUM SIZE OF DRIVING PULLEYS

Width of belt Inches	Diameter of driving pulley Inches	Width of belt Inches	Diameter of driving pulley Inches
12	16 to 18	26	24 to 30
14	16 to 18	28	24 to 30
16	20 to 24	30	30 to 36
18	20 to 24	32	30 to 36
20	20 to 24	34	30 to 42
22	20 to 30	36	30 to 48
24	24 to 30		

42 Only two or three sizes of pulleys are required for each width of belt to make up drives for conveyors of the various lengths. A short conveyor may be driven by a single bare pulley; a longer con-

veyor may be driven by a pulley of the same diameter if it is covered with rubber. Still longer conveyors would use a two-pulley drive with bare pulleys. These in turn may be rubber covered, giving still greater tractive power. The faces of all pulleys should be at least 2 in. wider than the belt. At some point in the length of the conveyor, a pair of take-ups should be used to take up the stretch in the belt, and to adjust it.

43 Fig. 10 shows one type of head drive. It will be noted that the driving is done by the larger pulley, while the material is discharged over the smaller pulley, thus saving head room and giving a clean discharge. This photograph was taken before the chute was placed.

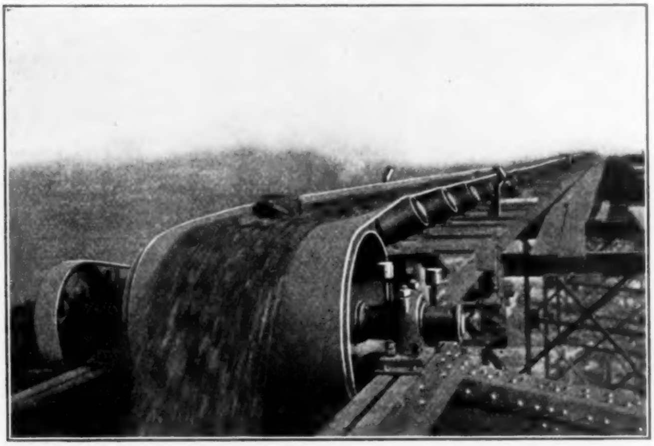


FIG. 11 DISCHARGE AT HEAD OF CONVEYOR

44 Fig. 11 is the discharge at head end of a conveyor. The drive is located at the foot of the incline on the ground, 300 ft. away. The pulley on the left drives the rotary brush. Capacity 600 tons per hour.

DISCHARGING DEVICES

45 Most conveyors receive material at one end or at various points in their length and discharge it over the head pulley. When necessary to discharge at points between the ends, either fixed or movable trippers are used. The fixed tripper consists of two pulleys, one located above and ahead of the other. The belt passes over the

upper pulley then over the lower in such manner that the material is discharged from the belt into a chute which will carry it to the side of the conveyor.

46 When fine sized material is being conveyed, these trippers may be made automatic, so that the stream of material will be deflected into the first bin until filled. The material will then fill up in the chutes, flow back on itself to the belt and be carried to the second tripper, and so on until all bins are filled. Should material be drawn from any bin, the stream will immediately be deflected until the bin is again filled and then pass on. The writer has used eight such trippers on a single conveyor.

47 The movable or traveling tripper is similar to the fixed tripper in form and operation. The two pulleys and chute are mounted

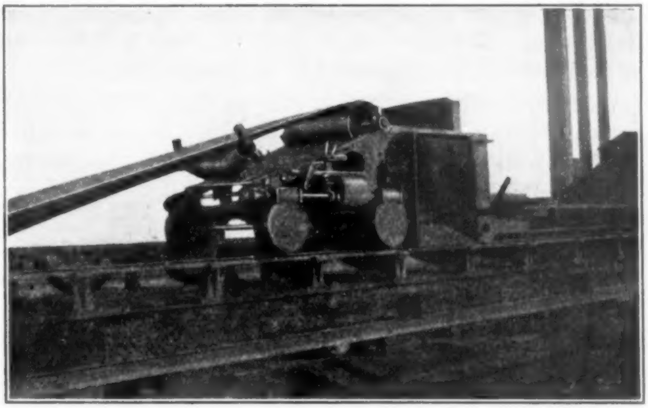


Fig. 12 Automatic Self-Reversing Tripper

on a frame supported on four flanged wheels. A pair of rails extending the length of the conveyor provides track for the machine and enables it to discharge material at any point in the length of its travel. The tripper is usually provided with some clamping device to secure it to the rails when discharging in a fixed position.

48 Trippers are moved in the following ways: The hand propelled type is moved by crank with pinion and gear to the truck wheels. Others use a wire rope with winch located at one end of the conveyor. The self propelled tripper is driven by power taken from the conveyor belt, through suitable gearing to the truck wheels. The automatic self reversing type is illustrated in Fig. 12.

49 A lever on one side properly connected with the driving mechanism controls the direction of travel of the machine. This lever engages adjustable stops located on the rails at the desired limits of discharge and the tripper will travel between these stops, automatically reversing at each end. A lever is provided which will throw this moving mechanism out of gear allowing the machine to discharge in a fixed position.

CLEANING BELTS

50 When the material handled is damp it is advisable to provide some mechanism to clean the belt after it passes over the discharge pulley; otherwise a small quantity will be carried back by the return belt. For this purpose, rotary brushes, made of various fibers, are used. These brushes revolve at a high speed, sweeping the material into the chutes. They are driven from the conveyor belts and are provided with means of adjustment for the wear of the fiber.

51 When very sticky material, such as clay, is being conveyed, a strong water spray is used to clean the belts. An air blast has been used to clean belts, but the large volume and high pressure required makes this method expensive.

CHUTES AND FEEDERS

52 One of the most important features of belt conveyor design is that of providing chutes which will properly load the material onto the belt. Mine run ore may be fed to a belt conveyor so that it will not injure the belt, but with improperly designed chutes the highest grade belt may be ruined in a few weeks when it should last for years.

53 It is impossible to lay down any rules for the design of chutes. It is a subject requiring a careful study of local conditions and a complete knowledge of the materials handled and of the manner in which they will move in chutes under various conditions.

54 The material should always be delivered to the belt in the direction of the belt travel and as nearly as possible at the same speed, so that it will go from chute to belt with no shock or jar. When feeding a conveyor from bulk, for example, from storage bin, some type of automatic feeder should be used to insure an even and continuous feed to the belt. This is particularly important when handling unsized material. Should a simple gate be used, it must be set for the large lumps, the result being a rush of fine material after the lump has

passed. If a feeder is so driven that it will start and stop with the conveyor it feeds, it will generally save the expense of an attendant at the loading end and prove more satisfactory.

55 The writer favors the shaking feeder consisting of an inclined pan set under the bin opening at such an angle that it stops the flow when stationary. When given a reciprocating motion by crank and connecting rod the material is moved along the pan to the belt. By varying the length of stroke and inclination of the pan the amount of material delivered may be absolutely regulated.

CAPACITY

56 Belt conveyors may be built to handle practically any quantity of material which may be fed to them. The following table gives the capacity, maximum size of lumps, and advisable speed for the differ-

TABLE 2 BELT CAPACITY AND SPEED

Width of belt	Maximum size of pieces	Capacity in cubic feet per hour at a belt speed of 100 ft. per min.	Capacity in tons per hour, at a belt speed of 100 ft. per min., material weighing 100 lb. per cu. ft.	Maximum advisable speed in feet per minute	Capacity in cubic feet at the maximum advisable belt speed	Capacity in tons per hour at the maximum advisable belt speed material weighing 100 lb. per cu. ft.
12	2	460	23	300	1380	69
14	2½	630	31	300	1890	94
16	3	820	41	300	2460	123
18	4	1040	52	350	3640	182
20	5	1280	64	350	4480	224
22	6	1550	78	400	6200	310
24	8	1850	93	400	7400	370
26	9	2180	110	450	9810	490
28	12	2500	125	450	11250	563
30	14	2900	145	450	13050	653
32	15	3300	165	500	16500	825
34	16	3700	185	500	18500	925
36	18	4200	210	500	21000	1050
38	19	4600	230	550	25300	1265
40	20	5100	255	550	28050	1402
42	20	5600	280	550	30800	1540
44	22	6200	310	600	37200	1860
46	22	6800	340	600	40800	2040
48	24	7400	370	600	44400	2220

ent widths of belts. This table is based on an even and continuous flow of material to the conveyor and in choosing width and speed, a full knowledge of the local operating conditions and character of the material, is necessary so that the table may be used with judgment.

SPEED AND SIZE OF BELTS

57 When the quantity to be conveyed is small, and the pieces large, the size of the material fixes the width of the belt and *the speed should be as low as possible to carry safely the desired load.*

58 When the quantity is great, the capacity fixes the width and in this case also *the speed should be as slow as possible.* A slow speed belt may be loaded more deeply than one at high speed and when a narrow belt is run much above the advisable speed, the load thins out and the capacity does not increase as the speed.

59 The maximum length of the different widths of conveyors is determined by the fiber stress in the belt and is therefore closely related to the load and speed. Naturally level conveyors may be built longer than those lifting material. Conveyors 1000 ft. from center to center handling 400 tons per hour have been most satisfactorily operated.

60 Another important factor in the design of conveyors at high speed, handling large quantities, is the flow of material in the chutes. A 36 in. conveyor handling 750 tons of coal per hour with a belt speed of 750 ft. per minute under a 10 000 ton pocket could not be loaded from a single chute, because it was not possible for the coal to attain a speed of 750 ft. per minute in the chute. It was necessary, therefore, in order to obtain a full load to open seven gates, each placing a layer of coal on the belt until the desired load was obtained. During a test this belt carried about 800 tons per hour.

POWER REQUIRED

61 The power required to drive a belt conveyor depends on a great variety of conditions such as the spacing of idlers, type of drive, thickness of belt, etc.

62 In figuring the power required, it is important to remember that the belt should be run no faster than is required to carry the desired load. If for any reason it is necessary to increase the speed, the figure taken for load should be increased in proportion and the power figured accordingly. *In other words, the power should always be figured for the full capacity at the chosen speed,* as follows:

C = Power constant from table.
T = Load in tons per hour.
L = Length of conveyor between centers in feet.
H = Vertical height in feet that material is lifted.
S = Belt speed in feet per minute.
B = Width of belt in inches.

For level conveyors,

$$\text{H. p.} = \frac{C \times T \times L}{1000}$$

For incline conveyors

$$\text{H. p.} = \frac{C \times T \times L}{1000} + \frac{T \times H}{1000}$$

Add for each movable or fixed tripper horse power in column 3 of table.

Add 20 per cent to horse power for each conveyor under 50 ft. in length.

Add 10 per cent to horse power for each conveyor beteeen 50 ft. and 100 ft. in length.

The above figures do not include gear friction, should the conveyor be driven by gears.

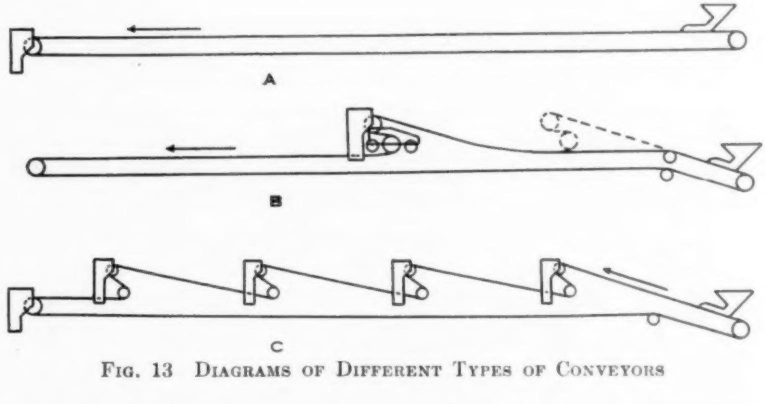
TABLE 3 POWER REQUIRED FOR GIVEN LOAD

	1	2	3	4	5
Width of belt	C for material weighing from 25 lb. to 75 lb. per cu. ft.	C for material weighing from 75 lb. to 125 lb. per cu. ft.	H. P. required for each movable or fixed tripper	Minimum plies of belt	Maximum plies of belt
12	.234	.147	½	3	4
14	.226	.143	½	3	4
16	.220	.140	¾	4	5
18	.209	.138	1	4	5
20	.205	.136	1¼	4	6
22	.199	.133	1½	5	6
24	.195	.131	1¾	5	7
26	.187	.127	2	5	7
28	.175	.121	2¼	5	8
30	.167	.117	2½	6	8
32	.163	.115	2¾	6	9
34	.161	.114	3	6	10
36	.157	.112	3¼	6	10

63 With the load and the size of material known, choose from the capacity table the proper width of belt and proper speed. The above formulae give the horse power required for the conveyor when handling the given load at the proper speed. With the horse power and the speed known, the stress in the belt should be figured by the following formula in order to find the proper number of plies. Stress in belt in pounds per inch of width $= \dfrac{\text{h. p.} \times 33\,000}{S \times B}$ With this value known, the number of plies may be determined, using 20 lb. per inch per ply as the maximum. The columns four and five of Table 3 give the maximum and minimum advisable plies of the different widths of belt. Belts between these limits will trough properly and will be stiff enough to support the load. The maximum number of plies determines the maximum length for each width of conveyor.

ARRANGEMENT OF CONVEYORS

64 Fig. 13, A, shows in outline the simplest form of level conveyor receiving material at one or more points and discharging over the end pulley.



A

B

C

FIG. 13 DIAGRAMS OF DIFFERENT TYPES OF CONVEYORS

65 In Fig. 13, B, the material is received at one end and discharged at any point in the length by a movable tripper. The tail end is usually depressed as shown so that the belt will not be raised against the loading chute by the tripper when discharging near the tail end as shown in dotted lines. Fig. 13, C, is a level conveyor with a series of fixed dumps.

66 Fig. 14, A, shows the simple inclined conveyor. Practically any material may be carried up 20 deg. to the horizontal when run at proper speed and with correctly designed chutes. With sized material, such as sand, crushed ore, stone or coal, the angle may be increased to 22 deg. and under some conditions, even more. When elevating material containing large lumps, the lumps have a tendency to roll back on the layer of fines so that the angle should not be over 20 deg. When handling round material, such as cement clinkers from rotary kilns, the angle should be from 12 deg. to 15 deg. maximum. Inclined conveyors should not be run at high speeds, as the material must be given the same speed as the belt and the higher the speed the more the slip at the loading point and the greater the wear on the belt.

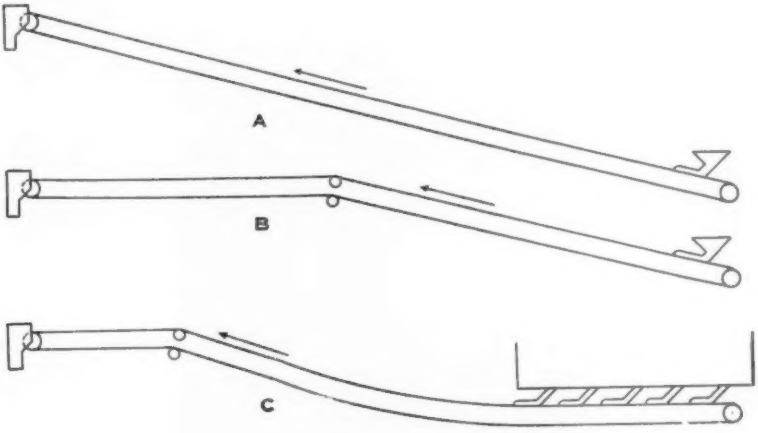


Fig. 14 Diagrams of Inclined Conveyors

67 Fig. 14, B, is the combination of inclined and level conveyor which is largely used. Where the belt passes from the incline to the level it should not pass over a troughing idler but over a high crowned pulley. This flattens the belt out at the bend and the material will leave the belt on a trajectory and land on the level portion with no spill, because belt and material are moving at the same speed.

68 It is frequently advio rsable tun conveyors on a vertical curve. Fig. 14, C, illustrates the combination of curved and level conveyor. The radius of the curve depends on the size of the belt, location of the drive and the local conditions. It is possible so to design these

conveyors that the belt will touch the troughing idlers, whether loaded or light. Should the belt lift off the idlers on the curve, it reduces the driving tension on the belt, which should be avoided. Fig. 15 is a photograph of the lower end of a conveyor on a curve.

69 Fig. 16 is a photograph showing the discharge from a level conveyor to the inclined portion of another, and illustrates the method of making the bend and the action of the material at this point.

70 Fig. 17, A, shows the combination of level conveyor, fixed dump, inclined conveyor and a number of fixed trippers. Frequently

Fig. 15 Lower End of Conveyor on Curve

it is not possible to obtain the room required for the curve belts, so the fixed dump is used. The material is discharged from the upper pulley of the dump into a chute feeding the inclined portion of the same belt. The fixed and movable trippers may be used with any of the above combinations where a portion of the conveyor is level or not inclined over 7 deg.

71 Fig. 17, B, is a diagram of still another combination, in which the conveyor starts level, carries material down hill with an incline and curve and then runs level again. Fig. 18 is a photograph of such a conveyor receiving material from the hoisting towers in the distance

Fig. 16 Discharge from Level to Inclined Conveyor

and running down hill in a concrete passageway under railroad tracks and highway.

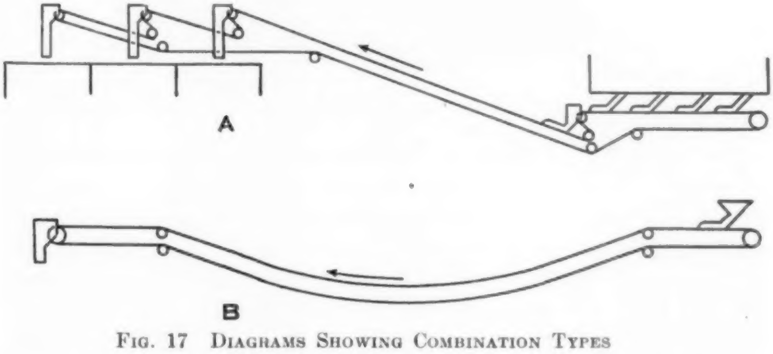


Fig. 17 Diagrams Showing Combination Types

72 Fig. 19 is a photograph of the conveyor dredge used to deepen Fox River, Wisconsin. The endless chain of buckets dig the mud, clay,

rock, etc., from the river bed, discharging to a 32 in. conveyor carried by the dredge. Three other conveyors mounted on scows carry the material to either shore, as may be desired, with but the one handling. The conveyors first used on this dredge were not satisfactory, owing to incorrect design, and they were rebuilt last year, one conveyor being equipped with rubber belt and three with Balata Belt. The latter proved more satisfactory, owing to its water proof qualities.

73 Fig. 20 illustrates a large coal storage plant. The coal is hoisted from vessels by the towers on the dock and fed to a 36 in. con-

FIG. 18 CONVEYOR RUNNING IN CONCRETE SUB-PASSAGEWAY

veyor about 600 ft. long back of the hoisting towers. This conveyor discharges to one running along the end of the pile, about 300 ft. in length, discharging to the tail of the inclined conveyor about 500 ft. long, to the left of the photograph. A 50-ft. conveyor carries the material from the top of the incline to the center of the pivot point of the bridge. A chute at this point feeds the 500 ft. tripper line carried by the bridge. The bridge swings through 180 deg. and the travel of the bridge and the movement of the tripper the length of the bridge conveyor makes it possible to receive coal at any point along the dock and

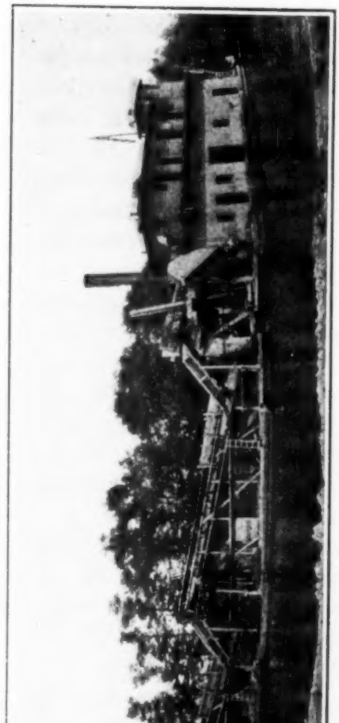


FIG. 19 CONVEYOR DREDGE

FIG. 20 COAL STORAGE PLANT

store it at any desired location in the storage space. The first two conveyors are driven from their head ends. The inclined belt is driven at the foot of the incline. The short belt at the top is driven by gears and chain from the head of the conveyor feeding it. The bridge conveyor is driven from the tail or loading end. All the motors are so wired that they may be stopped from any point along the line by means of push buttons. This system has a capacity of about 700 tons of mine run coal per hour and has handled in the past three seasons nearly one and a quarter million tons.

74 This paper has been presented mainly to acquaint the engineers who are designing plants with the possibilities of the belt conveyor for handling heavy abrasive materials, and to give them some data which will be of service in the preparation of preliminary designs. No one type of conveyor is perfect for handling every material. Each has its field, but it may be fairly said that it has become universal practice to use belt conveyors in places where ten years ago only chains were considered.

75 The ancient idea that only conveyors made of iron or steel could handle coal, ore, stone, etc., has been exploded and the following is a list of the materials that belt conveyors are most satisfactorily handling:

Coal in all sizes at the mine and in industrial plants.
Stone in crushing plants, etc.
Sand and gravel in washing plants, etc.
Concrete materials in mixing plants.
Mixed concrete from mixers to forms.
Ore, both run of mine and crushed in concentrating plants, crushing plants, reduction works, etc.
Earth, rock, clay, etc., in dredging and excavating operations.
Cement rock and other materials in cement mills.
Wood chips and pulp in pulp mills.
Small packages in shipping rooms of stores, and express depots.
Salt and chemicals.

76 The writer does not claim that the belt conveyors are the panacea for all the ills of the conveyor user, but when properly designed and installed, they fully justify the following claims:

a Large capacity with low power requirements.

b Small maintenance charges. Belts will last from 3 to 8 years dependent on the duty. Idlers and drives 10 to 15 years.

c Freedom from shut down, as there are no links to break and a belt will give months of warning before giving out.

d Light weight, resulting in lighter structures and saving in freight, particularly in ocean shipment and at isolated mining plants.

e Complete separation of the material carried from the moving parts—the material coming in contact only with the belt.

f Perfect alignment is not absolutely necessary as is the case with most other conveyors. The long incline conveyor to the left of Fig. 20 was built on a swamp. The two bents near the head tower shifted 10 in. and the conveyor was operated in this condition for nearly one month with no trouble or damage to the apparatus.

g Owing to light weight and the fact that perfect alignment is not necessary, they may be made up in portable sections, which are in great demand.

h Large overload capacities.

77 The above claims are based not on a few months' research, but on years of service as a mechanical engineer with one of the large anthracite coal mining companies and over eight years devoted entirely to the design, manufacture and installation of belt conveyors, the early years being pioneer work when the industry was in its infancy.

78 In choosing illustrations for this paper, preference has been given to the line drawings and photographs illustrating parts of conveyors rather than those which show complete installations, as the object has been, not to describe existing plants, but to bring out the principles of the mechanism. Acknowledgment is due the companies who have consented to the use of photographs taken at their plants.

NOTE.—The discussion on this paper and the author's closure are published under Discussion on the Conveying of Materials, No. 1195, in this volume.—THE EDITOR.

No. 1193

CONVEYING MACHINERY IN A PORTLAND CEMENT PLANT

By C. J. Tomlinson, Neponset, Ill.

Non-Member

In the general expansion of industries experienced during the past few years, the larger works have presented a favorable field for the increased use of the labor saving devices for the transportation of materials. These devices are to a large extent machines that have been in use in some industries for a number of years, and in these locations have given long and efficient service. But under the new conditions, operating under increased capacities, with heavier or more exacting duties, inherent defects develop that have been of small consequence in their former uses; and it is expedient to study this machinery with a view to deciding on a more economical or intelligent use of it.

2 The problem of the proper form of conveying machinery is often of most fundamental importance, preliminary to the design of works involving, in a measure, the cost and arrangement of buildings and the care, repair, and use of the machinery served.

3 There are a number of conveying devices in use in certain industries that have apparently given most satisfactory results, and it is therefore natural for one to resort to them for use in works that involve the conveying or elevating of quantities of loose materials. The devices referred to are: the belt conveyor, the screw-conveyor, the chain and bucket elevator, and the chain and bucket conveyor carrying or dragging the material with either buckets or flights.

4 This machinery is being tested out in the portland cement plant under conditions that are of utmost interest to engineers, involving as they do the question of the economical limit of their use.

5 Of these devices, the belt conveyor occupies a leading place, and it has given good results when handling grain or any material

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers,

not heated, sticky, or light enough to be objectionably dusty. It makes a convenient conveyor for contractors' use as it does not require expensive supports, or much care in their erection.

6 When used in permanent constructions it should be erected on well supported rails, elevated enough above the floor to allow for convenient cleaning up under it. In finely powdered materials the amount carried over and dislodged from the face of the belt at the leading return rolls will make it desirable for these rolls to be elevated enough above the floor to allow a noticeable heap to accumulate, before interfering with the rolls.

7 Where the conveyor is used over bins, as in cement stock houses, it is often arranged without a floor beneath the belt and with the oiler's walkway composed of a number of narrow planks, spaced enough apart to prevent a great accumulation of the spill on them. This is a somewhat slovenly arrangement, but it is such a disagreeable place to work in these galleries, owing to the dust from the tripper, that it is expedient to prevent the necessity of cleaning up as much as possible.

8 If the carrying rolls of a conveyor are properly set, guide rolls will not be needed, and, as the belt is the largest item of expense, care should be taken to prevent undue straining of it around small pulleys, by using high angle troughing rolls, or by placing them close to a foot or head pulley.

9 In the arrangement of spouts or tripper boxes, a study of the characteristics of the material carried should be made. Occasionally it occurs, as is the case with the raw material used in cement making, that it is dry and dusty; and again damp and sticky; and a spout that will deliver the damp material will be unduly destructive on the belt when the material is running dry. A pocket of the material, provided to take the wear in an angle in a spout or in a box, is particularly liable to give trouble in a sticky material.

10 The maximum angle at which a belt conveyor will elevate is largely determined by the angle of repose of the material, and also somewhat by the nature of the same; as for instance, a material containing a large percentage of spherical bodies having a tendency to roll. In order to prevent the slight jar imparted to the material through a belt by the carrying rolls, they should be spaced considerably closer on a steep incline than would be necessary on the level; and care should be taken to insure their being in good alignment. If large diameter return rolls are used, they may be spaced farther apart than has been usual, and a distance of 20 to 25 ft. is not too great.

11 When handling a material that is very dusty, or is explosive, as for instance powdered coal, or where the head room is much restricted, the screw conveyor has given good results; but on account of the weight and grittiness of the materials handled in portland cement work, it does not give the satisfaction it has given in other industries. It has the inherent defect that it shoves or drags the material and does not carry it, and is essentially a special machine, to be resorted to only when it is evident that other conveyors can not be used. The chain and bucket elevator, using a single heavy bushed chain and overlapping or continuous buckets, has been used for crushed rock, coal and clinker. It is usually a slow speed machine having a chain speed of 80 to 120 ft. per minute. The high speed centrifugal discharge elevator is more suited for lighter materials, or for contractors' use, although it has been used considerably in powdered materials or in cement.

12 When arranging the drives for several elevators and conveyors, as in a coal grinding room where storage bins are provided over the machines, a simple arrangement of light drives from a line shaft is impracticable. This is especially true if it is wished to take advantage of the storage space, in preventing a general shut down in case of an accident to the machinery. A study of the requirements raises the question as to the advisability of placing bins below as well as above this machinery.

13 The large amount of dust in the air of a grinding room often causes belts or ropes to give trouble. The slow speed of revolution of the various head shafts is also a factor in the simple arrangement of motors, and as far as possible it is expedient to isolate the latter in motor rooms. The small air gap of induction motors makes them somewhat unsafe in the presence of so much gritty dust, while the models with larger air gaps are expensive.

14 The use of the pivoted bucket conveyor often simplifies the arrangement of drives. These machines, as ordinarily built, can be fed, or will deliver, at any point along the upper or lower runs, and will also elevate perpendicularly, combining in one machine the elevators and conveyors required to serve a number of machines. The long links—usually 18 in.—eliminate some of the difficulties due to wear and uneven pitch experienced in the ordinary chain and sprocket drive. It is not practicable to enclose this machine in casings for use in powdered materials.

15 This type has found increased favor among manufacturers of cement, owing to its ability to handle heated material, although it has signally failed to give satisfaction in handling a product as highly heated as the clinker falling directly from the kilns.

16 The devices mentioned are continuous and automatic in operation, but present several limitations in their design. They are composed of a number of small parts and journals so placed as to be dependent on the failure of any one part.

17 They do not incorporate a means for measuring the production of each of a number of machines being served, nor will they serve in preparing a compound of materials, as in a location under a series of bins, from any of which it is desired to draw a certain proportion of material.

18 The small trucks are excessively heavy for the load they carry, and have a mechanical efficiency as proportionately low as that of the modern large size freight car is high. On account of the conditions imposed in cement manufacture I have offered reasons for advising the use of heavy skip hoists and scale, transfer cars, similar to those in use in blast furnace work.[1]

19 Such devices require buildings of increased cost, owing to the height necessary to provide storage space beneath and over the machines; and also the elevation of these bins to admit the large scale car beneath them.

20 In return they offer data for comparing performances of a number of types of machines, records of the different shifts of operatives, and an interchangeable and flexible conveying equipment that is a great aid in repairing and maintaining heavy grinding machinery that is required to run continuously, for an indefinite period.

21 As they are non-automatic they will not compare favorably with their automatic rivals when handling a moderate tonnage; but where a given system is called upon to handle 30 to 40 tons per hour and over, the smaller machines require attendants enough to balance the increased attention required by the scale cars, and the mechanical efficiency of the devices becomes a controlling factor.

NOTE—The discussion on this paper and the author's closure are published under "Discussion on the Conveying of Materials", No. 1195 in this Volume.
—EDITOR.

[1]*Engineering News*, February 13, 1908.

No. 1194

PERFORMANCE OF BELT CONVEYORS

By Edwin J. Haddock, Columbus, Ohio

Member of the Society

Of the half dozen forms of conveyors commonly used for handling material in bulk, or for continuous conveying, one of well deserved and increasing popularity on account of its capacity, economy of power, noiselessness, and gentle handling of its load, is the belt conveyor.

2 This form of conveyor is a very efficient means of conveying coal, crushed ores, grain, small goods in packages, etc., when the transportation is in a straight line, either horizontal, or inclined at an angle not exceeding, in good practice, 20 deg.

3 Some experiments with belt conveyors, made some time ago by the company with whom the writer is connected, may be of interest. Data relating to the following points were desired:

a Other conditions remaining the same, what will be the effect on the traction of changing the diameter of the driving pulley?

b What is the effect of different arcs of contact on the tractive force exerted by the driving pulley?

c What is the effect of different initial tensions of the belt on the tractive force?

d What is the value of rubber covered pulleys as compared with plain ones?

4 An experimental belt conveyor was constructed as shown by Fig. 1 and Fig. 2. This conveyor was driven by a 220 volt d.c. motor, and the driving machinery was calibrated for torque as shown in Fig. 3 and Fig. 4.

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

5 A prony brake was placed on the head shaft of the conveyor (the conveyor belt having been removed) and a pair of platform scales

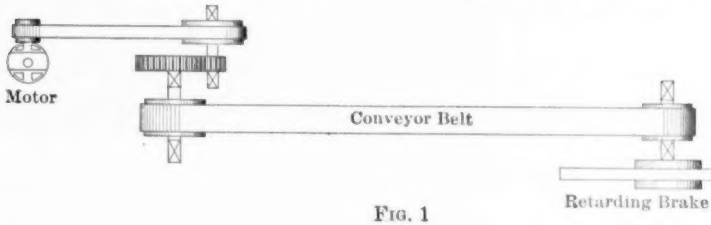


Fig. 1

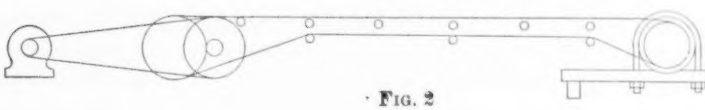


Fig. 2

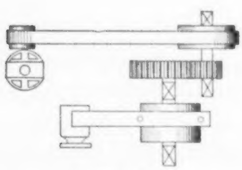


Fig. 3

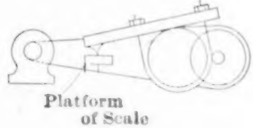


Fig. 4

Experimental Conveyor and Brake for Measuring Torque

placed under it, as shown in the illustrations. A d.c. voltmeter and ammeter were placed in the electric circuit, and the readings, shown in Table 1, taken, from which the calibration curves, Fig. 5, were constructed, by which the torque on the head shaft may be determined from any reading of the ammeter, the voltage being kept at 220.

TABLE I CALIBRATION OF DRIVING GEAR

Scales load	Torque 12 in. radius	Amperes, average four observations	Volts, average four observations	Watts average four observations	Amperes at 220 volts
0	0	5.0	222.0	1110	5.05
25	225	9.87	225.7	2229	10.13
50	450	14.25	222.5	3170	14.41
75	675	18.87	220.0	4152	18.87
100	900	22.75	218.0	4960	22.54
125	1125	27.25	216.5	5898	26.81
150	1350	32.62	218.0	7112	32.34
175	1575	37.50	213.0	7996	36.35
200	1800	42.00	214.0	8988	40.85

6 A 12 in. 4-ply conveyor belt, 158 ft. long, was placed on the conveyor, a joint being made by lapping and riveting the ends. The total weight of the belt was 380 lb., or 2.405 lb. per foot. The retarding brake was placed at the foot shaft of the conveyor.

7 Experiment 1 was made to determine the effect of changing the diameter of the driving pulley. The tension was kept constant by

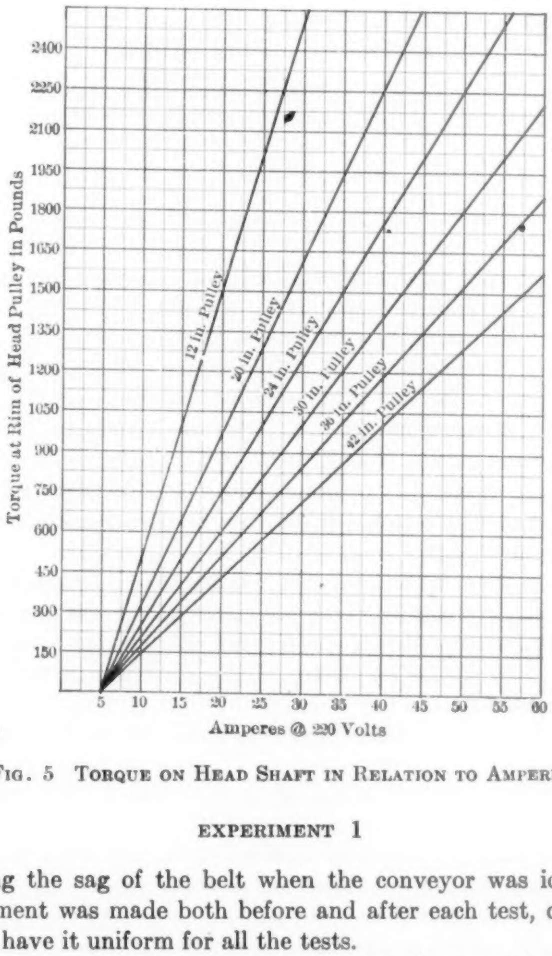


Fig. 5 Torque on Head Shaft in Relation to Amperes

EXPERIMENT 1

measuring the sag of the belt when the conveyor was idle. This measurement was made both before and after each test, care being taken to have it uniform for all the tests.

8 In all the tests the brake on the foot shaft was tightened until the belt slipped on the head pulley, and the reading was taken at that

moment. An average of at least four readings is given in the table. In each case, the ammeter read one to three amperes higher just before the belt slipped than it did as the belt was slipping, the hand dropping back to the reading again.

9 The observations shown in Table 2 show the effect of different diameters of the driving pulley.

TABLE 2 EFFECT OF CHANGE IN DIAMETER OF DRIVING PULLEYS

Order of test	Amperes at 220 v.	Diameter of pulley in inches	Belt slipped at lb
4	12	12	720
3	19	20	855
2	22	24	855
1	27	30	895
5	32	36	910
6	35½	42	885

10 The initial tension of the belt was not measured in pounds, care only being taken to keep it uniform as above described.

11 The belt was new and covered with the usual preservative mineral powder, which was wiped off from time to time. This circumstance, together with any possible errors of tension, would account for most of the discrepancies shown. The results clearly show that the diameter of the pulleys from 42 in. to 20 in. had no appreciable effect on the belt in question (4-ply), but that a decided falling off occurred in the 12 in., indicating that this pulley was too small to run with a belt of this thickness.

12 From these data the writer assumes that if a pulley is not less than 5 in. in diameter for every ply of the belt it will exert its maximum tractive force, the excess diameter above this being simply a matter of convenience for drive. If the diameter is reduced much below this amount, the belt will not adhere closely to the pulley, and therefore the complete benefit of the pressure will not be realized.

EXPERIMENT 2

13 The second experiment was made to determine the effect of different arcs of contact. The conveyor was arranged as in Experiment 1, but considerable difficulty was met in getting the tension uniform, and very erratic results were obtained.

14 The foot shaft was then rearranged, as shown in Fig. 6, the driving machinery at the head of the conveyor being used without

any change, except that the snubbing idlers were modified to get the required arc of contact.

15 With a 42 in. pulley on the head shaft, a 1000 lb. tension weight at the foot shaft, the following results were obtained, as shown in Table 3.

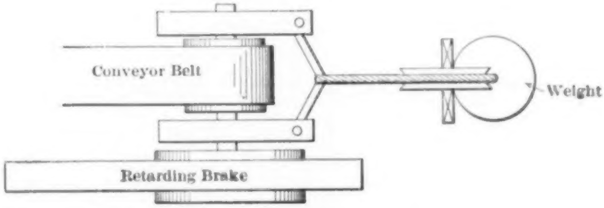


FIG. 6 FOOT SHAFT ARRANGEMENT

TABLE 3 EFFECT OF DIFFERENT ARCS OF CONTACT

Arcs of contact	Amperes	Effective pull on belt in pounds	Load 180 deg. = 100
1/6 or 60 deg.	11	180	32
1/3 or 120 deg.	19	405	73
1/2 or 180 deg.	24	550	100
5/8 or 225 deg.	32	775	141
2/3 or 240 deg	34	840	153

16 From this table, the curve shown in Fig. 7 was constructed. It will be noticed that below the 180 deg. contact, the curve is fairly a straight line, but above this point there seems to be a curve in which the traction increases at a more rapid rate. It is unfortunate that more observations were not taken from the 180 deg. line up to get the exact form of this curve. Making the curve a straight line, however, throws the error on the safe side.

EXPERIMENT 3

17 Experiment 3 was made to determine the effect of initial tension on the tractive force: With the same arrangement of conveyor as in Experiment 2, including the 42 in. pulley on the head shaft, and the snubbing idler arranged to give a contact of 180 deg. on the drive pulley, different tension weights were hung on the foot of the conveyor and the readings shown in Table 4 were obtained.

18 The tests are given in the order made. When the 2000 lb. weight was applied, the belt stretched abnormally from its length

under the 1500 lb. weight, the additional 500 lb. causing the belt to elongate about 3 ft. in the total length of 158 ft. The belt then seemed to stop stretching, and remained at a constant length until the load was removed. As the weight was reduced gradually to 750 lb., the belt recovered 15 in. of this stretch, while the weight of 3000 lb. caused the belt to stretch 8 in. longer than the 2000 lb. weight had

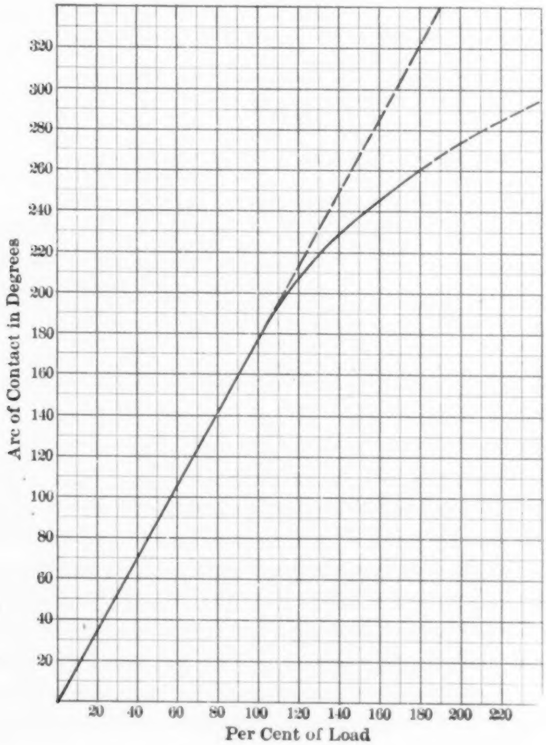


FIG. 7 CURVE PLOTTED FROM TABLE 3

done. The belt was then run under this tension for a period of 30 minutes without changing its length to any appreciable degree. The belt strain of 1500 lb., caused by the application of a 3000 lb. weight, is equivalent to a stress of 31.25 lb. per inch per ply. No reason appears against using 40 lb. per inch width per ply (the manufacturers' guarantee) as to the ultimate working strength of any rubber con-

veyor belt. This is the maximum value which should be allowed in the general equations following.

19 From Table 4 Fig. 8 was constructed, which shows the various belt strains when the arc of contact is 180 deg., and the initial tension on the belt is known.

TABLE 4 EFFECT OF DIFFERENT TENSIONS ON THE SAME PULLEY

Tension weight	Initial belt tension	Amperes	Effective pull on belt	Belt strain	
				Maximum	Minimum
650	325	14.0	260	455	195
800	400	16.5	330	565	235
1300	650	24.0	550	925	375
1500	750	29.0	690	1095	405
2000	1000	38.0	950	1475	525
1500	750	31.0	750	1125	375
1250	625	25.0	575	913	337
1000	500	22.0	490	745	255
750	375	18.0	370	560	190
3000	1500	56.0	1450	2225	775

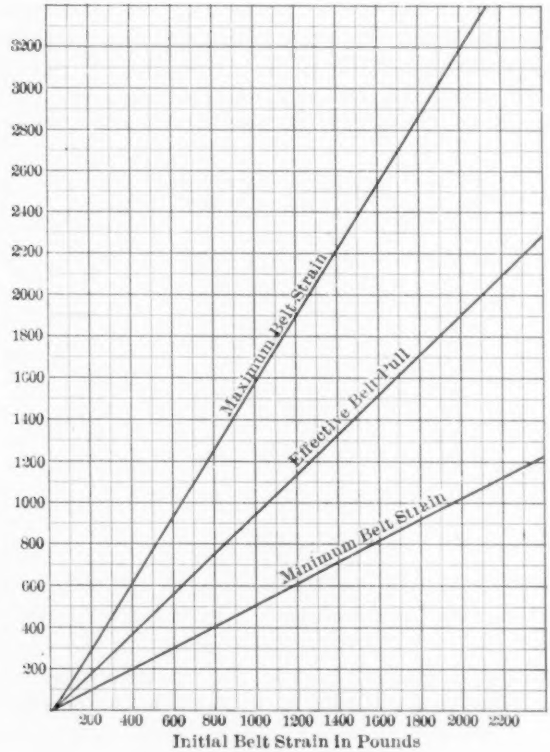
EXPERIMENT 4

20 Experiment 4 was made to determine the relative value of rubber faced vs. iron faced pulleys. A 24 in. rubber covered pulley was compared with a 24 in. iron faced pulley, on the experimental conveyor, with 180 deg. arc of contact, and a tension weight of 2000 lb. throwing an initial strain of 1000 lb. on the belt, the results shown in Table 5 were obtained, or 7 per cent in favor of the rubber faced pulley when the belt was clean and dry.

TABLE 5 EFFECT OF RUBBER FACING ON PULLEYS

	Amperes	Traction pounds	Per cent
24 in. iron faced pulleys....................	24	960	100.0
24 in. rubber faced pulleys.................	25½	1035	107.8

21 This experiment was performed at two different times, with an interval of several weeks between, and with substantially the same results. In order to carry it a little further, the device shown in Fig. 9 was provided to determine the effect of different conditions of the belt surface on rubber and iron faced pulleys.



G. 8 PLOTTED FROM TABLE 4

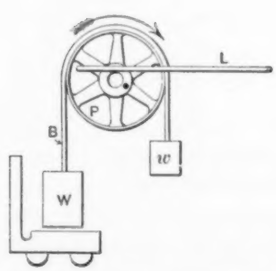


FIG. 9 APPARATUS TO TEST FRICTION ON PULLEYS

22 In this figure, W represents a heavy weight on the platform scales; B is a 6 in. 3-ply rubber belt; P is a 24 in. iron or rubber faced pulley; w is a small tension weight; and L is a hand lever by which the pulley was rotated. When the pulley was rotated slowly by hand, in the direction of the arrow, the scales would indicate the amount of weight not lifted by the traction of the belt, and hence the belt traction could be computed. The results are shown in Tables 6, 7 and 8, which also show the effect of foreign material between the belt and pulley.

23 When foreign material was introduced between the belt and the pulley it seemed to crush in between the two surfaces, so that if it was of a gritty or abrasive nature, it increased the tractive force, while a material which acted more or less as a lubricant, diminished that force. The abrasive material scratched the belt very much, and gave evidence of shortening its life.

24 From the foregoing experiments, the following general equations for traction may be deduced:

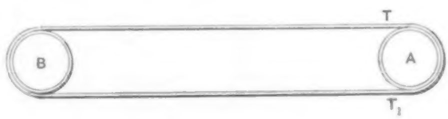


Fig. 10 Conveyor Diagram

THEORETICAL BELT CONVEYOR

Let Fig. 10 represent such a conveyor
- A, the driving or head pulley
- B, the driven or foot pulley
- T, the maximum pull on belt
- T_1, the minimum pull on belt
- F, the tractive force exerted by the pulley A
- T_2, the initial tension of belt
- ι the portion of the pulley in contact with the belt

$$= \frac{\text{arc of contact in degrees}}{360}$$

25 c is a constant depending on the arc of contact $= 2$ from 0 to 180 deg., and about 2.25 at 240 deg. For practical purposes, c may be taken at 2 for all arcs and the errors will be on the safe side.

26 k is a constant depending on the condition of the belt $= 1$ for good dry belts, but will vary from 40 per cent to 200 per cent of this value under different conditions. (See Tables 6, 7 and 8.)

TABLE 6 IRON FACED PULLEY

	Scales load $W = 227$	Net weight lifted lb.	Counter weight w lb.	Traction lb.
Clean dry belt	22	205	52	153
Clean damp belt	34	193	52	141
Dry coal dust	33	194	52	142
Damp coal dust	58	169	52	117
Dry clay	78	149	52	97
Damp clay	90	137	52	85
Dry slate	79	148	52	96
Damp slate	45	132	52	130
Dry sharp sand	110	117	52	65
Damp sharp sand	80	147	52	95

TABLE 7 RUBBER FACED PULLEY

	Scales load $W = 327$	Net weight lifted lb.	Counter weight w lb.	Traction lb.
Clean dry belt	109	218	52	166
Clean damp belt	120	207	52	155
Dry coal dust	152	175	52	123
Damp coal dust	195	132	52	80
Dry clay	190	137	52	85
Damp clay	175	152	52	100
Dry slate	63	264	52	212
Damp slate	50	277	52	225
Dry sharp sand	152	175	52	123
Damp sharp sand	152	175	52	123

TABLE 8 TRACTION OF IRON FACED VS. RUBBER FACED PULLEY

	Iron faced lb.	Rubber faced lb.	Iron faced per cent	Rubber faced per cent	Ratio Iron = 100
Clean dry belt	153	166	100	108	108
Clean damp belt	141	155	92	101	110
Dry coal dust	142	123	92	80	86
Damp coal dust	117	80	76	52	68
Dry clay	97	85	63	55	88
Damp clay	85	100	55	65	117
Dry slate	96	212	68	138	221
Damp slate	130	225	85	147	173
Dry sharp sand	65	123	42	80	190
Damp sharp sand	95	123	62	80	129

27 The general formulae for traction may be represented thus:

$$(a)\ F = i c k\, T_2 = T - T_1 = \frac{i c k}{2}(T + T_1) = \frac{2\, i c k\, T}{2 + i c k} = \frac{2\, i c k\, T_1}{2 - i c k}$$

$$(b)\ T_2 = \frac{T + T_1}{2} = \frac{F}{i c k} = \frac{2\, T}{2 + i c k} = \frac{2\, T_1}{2 - i c k}$$

$$(c)\ T = T_1\left(\frac{2 + i c k}{2 - i c k}\right) = T_2\left(\frac{2 + i c k}{2}\right) = F\left(\frac{2 + i c k}{2\, i c k}\right)$$

$$(d)\ T_1 = T\left(\frac{2 - i c k}{2 + i c k}\right) = T_2\left(\frac{2 - i c k}{2}\right) = F\left(\frac{2 - i c k}{2\, i c k}\right)$$

27 Calling $c = 2$ and $k = 1$, these formulae reduce to the following for clean dry rubber belts on iron pulleys:

$$(e)\ F = 2\, i\, T_2 = T - T_1 = i\,(T + T_1) = \frac{2\, i\, T}{1 + i} = \frac{2\, i\, T_1}{1 - i}$$

$$(f)\ T_2 = \frac{T + T_1}{2} = \frac{F}{2\, i} = \frac{T}{1 + i} = \frac{T_1}{1 - i}$$

$$(g)\ T = T_1\left(\frac{1 + i}{1 - i}\right) = T_2\,(1 + i) = F\left(\frac{1 + i}{2\, i}\right)$$

$$(h)\ T_1 = T\left(\frac{1 - i}{1 + i}\right) = T_2\,(1 - i) = F\left(\frac{1 - i}{2\, i}\right)$$

28 It should be borne in mind that T indicates the maximum belt strain, and is seldom, if ever, properly represented by F, which is the pull or tractive force, and is less than T.

29 The makers of conveyor belts will guarantee a working value of 40 lb. per inch per ply for T in rubber belting, and this should not be exceeded. Good practice would indicate a value of 30 lb. or less per inch per ply.

30 When multiple drive heads are used, the traction of each drive pulley should be computed separately, starting with the maximum belt strain T for the first pulley, computing T_1, and making T for the second pulley equal T_1 of the first, and so on, for the number of pulleys, the difference between the first tension and the last one will be the tractive force.

31 The following tables have been computed from the foregoing formulae, with the values as indicated by these experiments, for clean dry rubber belts on iron faced pulleys. Due allowance must be made for any other condition of belts. See Tables 6, 7 and 8.

TABLE 9

DATA CONCERNING IRON PULLEYS FOR DIFFERENT ARCS OF CONTACT FOR CLEAN DRY RUBBER BELTS WITH A MAX. BELT STRAIN OF 30 LB. PER IN. PER PLY

Degrees contact	Belt strain pounds		Initial tension T_2	Effective pull or traction F
	Max. T	Min. T_1		
100	30	17.0	23.5	13.0
110	30	16.0	23.0	14.0
120	30	15.0	22.5	15.0
130	30	14.3	22.2	15.9
140	30	13.2	21.6	16.8
150	30	12.3	21.2	17.7
160	30	11.5	20.8	18.5
170	30	10.7	20.4	19.3
180	30	10.0	20.0	20.0
190	30	9.3	19.7	20.7
200	30	8.6	19.3	21.4
210	30	7.9	19.0	22.1
220	30	7.2	18.6	22.8
230	30	6.6	18.3	23.4
240	30	6.0	16.0	24.0
250	30	5.4	17.7	24.6
260	30	4.8	17.4	25.2
270	30	4.3	17.2	25.7
280	30	3.8	16.9	26.2

32 The carrying capacity of belts is largely a question of condition of material. On the whole, a belt will carry a larger number of cubic feet of fine material, such as grain or crushed coal, or rock, than it will if the material consists largely of lumps, such as Run of Mine coal. In the following table of capacity, such fine material alone is considered, carried on a horizontal conveyor.

33 The horse power required to drive belt conveyors is too broad a subject to be fully covered in this paper. It varies so much with the design and spacing of the idlers, and the local conditions, that no accurate rules applicable to all cases can be given. The following formula may be found useful for the average case:

W = total weight of material delivered per minute
W^1 = total weight of belt
S = speed of belt in feet per minute
$H.P.$ = horse power
W_2 = maximum weight of material on belt at any one time.
H = total height to which material is elevated by conveyor

$$\text{h.p.} = \frac{(0.15\,W_1 + 0.07\,W_2)\,S + HW}{33\,000}$$

34 This will give the average h.p. In addition an allowance of from 4 per cent to 6 per cent should be made for each tripper, and all other unusual elements taken into consideration.

35 It is often desirable to change a belt conveyor at one or more points in its length, from horizontal to inclined, or to a greater incline by means of an upward curve, and it then becomes essential to know the curve assumed by an empty belt under such conditions in order to ascertain the minimum radius to which such a curve may be laid out, and the belt lay down on the idlers.

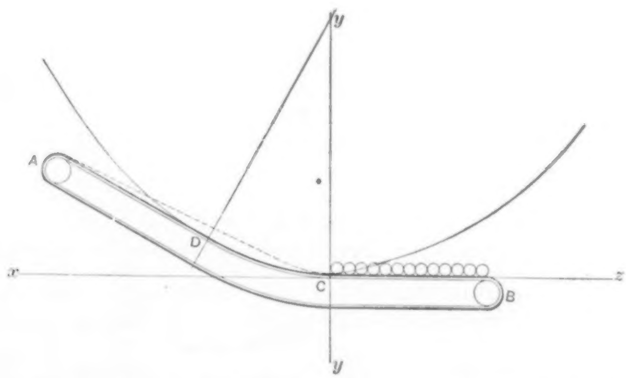


FIG. 11 DIAGRAM SHOWING CHANGE IN DIRECTION OF BELT

Let Fig. 11 represent such a condition in which
A = the head pulley
B = the tail pulley
C = the point where curvature begins
D = the point where curvature ends
XZ and YY = Rectangular coördinates with origin at C
T = Maximum pull on belt at point C
W = Weight of belt per foot of length

TABLE 10

DATA CONCERNING IRON PULLEYS FOR DIFFERENT ARCS OF CONTACT FOR CLEAN DRY RUBBER BELTS WITH MAXIMUM BELT STRAIN OF 100 PER CENT PER INCH PLY

Degrees contact	Belt strain per cent		Initial tension T_2 in per cent	Effective pull or traction F in per cent
	Max. T in per cent	Min. T_1 in per cent		
100	100	57	79	43
110	100	53	77	47
120	100	50	75	50
130	100	47	73	53
140	100	44	72	56
150	100	41	70	59
160	100	38	69	62
170	100	36	68	64
180	100	33	67	67
190	100	31	66	69
200	100	29	65	71
210	100	26	63	74
220	100	24	62	76
230	100	22	61	78
240	100	20	60	80
250	100	18	59	82
260	100	16	58	84
270	100	14	57	86
280	100	13	56	87

TABLE 11

SHOWING CROSS SECTION IN SQUARE FEET OF THE EFFECTIVE STREAM OF MATERIAL WHICH MAY SAFELY BE CARRIED BY BELT CONVEYORS

Width of belt	Effective cross section		Width of belt	Effective cross section	
	Flat belt	Troughed belt		Flat belt	Troughed belt
10	0.014	0.05	26	0.19	0.40
12	0.025	0.07	28	0.23	0.47
14	0.04	0.11	30	0.27	0.55
16	0.057	0.15	32	0.32	0.63
18	0.078	0.19	34	0.36	0.71
20	0.102	0.23	36	0.40	0.80
22	0.13	0.28	42	0.57	1.00
24	0.16	0.34	48	0.77	1.20

36 It is evident that the tendency of the belt to rise above the idlers on the curve as shown by dotted line, will be the greatest when the strain at C is greatest, and when the belt from C to D is lightest; that is, the tendency to rise will be greatest when the less inclined portion of the conveyor is loaded up to the point C and empty from C to D. Then according to the law of the catenary, the hanging belt will assume the curve of a parabola whose equation is

$$X^2 = \frac{2T}{W} Y$$

from which the curve may be plotted.

37 The belt will under the above maximum conditions never fall within the curve as computed from the above formula, between the points C and D, so that if the idlers are brought up to or a little above this curve, the belt will never leave the idlers.

38 When conditions permit, the radius of the curve C to D is usually made 300 ft. which is generally found satisfactory for all belts.

NOTE—The discussion on The Conveying of Materials is published under No. 1195 in this volume.—THE EDITOR.

No. 1195

THE CONVEYING OF MATERIALS

DISCUSSION

At the Detroit meeting the second session was devoted to a symposium on the hoisting and conveying of materials and five papers were presented as follows:

Hoisting and Conveying Machinery, G. E. Titcomb.
Continuous Conveying of Materials, S. B. Peck.
The Belt Conveyor, C. Kemble Baldwin.
Conveying Machinery in a Cement Plant, C. J. Tomlinson.
Performance of Belt Conveyors, E. J. Haddock.

Mr. Spencer Miller Mr. Titcomb briefly refers to cableways, showing that he regards such as hoisting and conveying machines. This is my excuse for adding to his paper something on the subject of cableways.

2 By a cableway I mean a hoisting and conveying machine employing a suspended cable as a track-way. When the hoisting function is omitted we call the device a wire rope tramway. There are about 1000 cableways in the United States. The field covered by the cableway is so extensive and the variety of cableways so great, that the subject is worthy of a separate paper. The loads handled by cableways range from one-half ton to 25 tons. Fifty-ton cableways have been designed, but so far as I know, have not been constructed. The length of the cableway span varies from 200 ft. to 2500 ft.

3 In the matter of speed, the load carriage travels usually from 600 to 1500 ft. per minute, while there are instances of cableways, especially those used in coaling warships at sea, where the carriage speed is over 3000 ft. per minute. At the Lidgerwood cableway testing station, speeds of 3600 ft. per min. have been accomplished and apparently such speeds are entirely practical for light loads. Cableways therefore seem to outdistance bridge tramways in the matter of speed and span. There are few problems solvable with the bridge tramway that are not solvable with a cableway. Cable-

ways have been built with "man trolleys" which eliminate the long hoisting and conveying ropes, with their attendant fall rope carriers.

4 Recent inventions properly applied multiply the life of the main cable and remove any objection on the standpoint of repairs. Cableways are successfully and economically employed in unloading ships at the wharf, and operating grab buckets and scraper buckets. For the construction of dams and locks they stand supreme. As excavators of canals they are frequently the most economical machine to employ. The most economical method of employing cableways in canal excavation is by the use of what is known as the duplex cableway. Two double towers placed on wheels are movable by power, lengthwise of the canal. Two cableways cross the canal. Grab buckets are handled by the cableways. Spoil banks are on both sides of the canal. Material is excavated by one cableway and carried speedily towards one tower, while by the other cableway it is carried in the opposite direction. Hence the bucket travels only half the distance, and therefore makes many more trips than a single cableway which has to do the entire work.

5 Many cableways are operated by electric motors both with the alternating and direct currents.

6 The greatest contest for economy of canal construction was witnessed in the excavation of the Chicago Drainage Canal, where the two leading machines were a bridge tramway and a traveling cableway. When all costs were taken into consideration, repairs, interest on plant, etc., the cableway won by several cents per cubic yard. It may be truthfully said that the cableway with its towers traveling on wheels was born on the Chicago Drainage Canal. This was also the first great construction work where the bridge tramway was employed. It would be interesting to know if a bridge tramway has been employed on construction work since. The great field of usefulness in bridge tramways has been in ship building, unloading ore and coal, and serving as a material handling machine in manufacturers' plants.

Mr. Melvin Pattison[1] In his paper on Hoisting and Conveying Machinery, Mr. Titcomb has given the type of ore-handling machinery that is most in use very little mention; and while he refers to some very interesting coal-handling equipments, he does not mention at all the type that ranks at least as high as any to which he refers.

2 In discussing Mr. Titcomb's paper, I do not find much to

[1] Melvin Pattison, Brown Hoisting Mchy. Co., Cleveland, O.

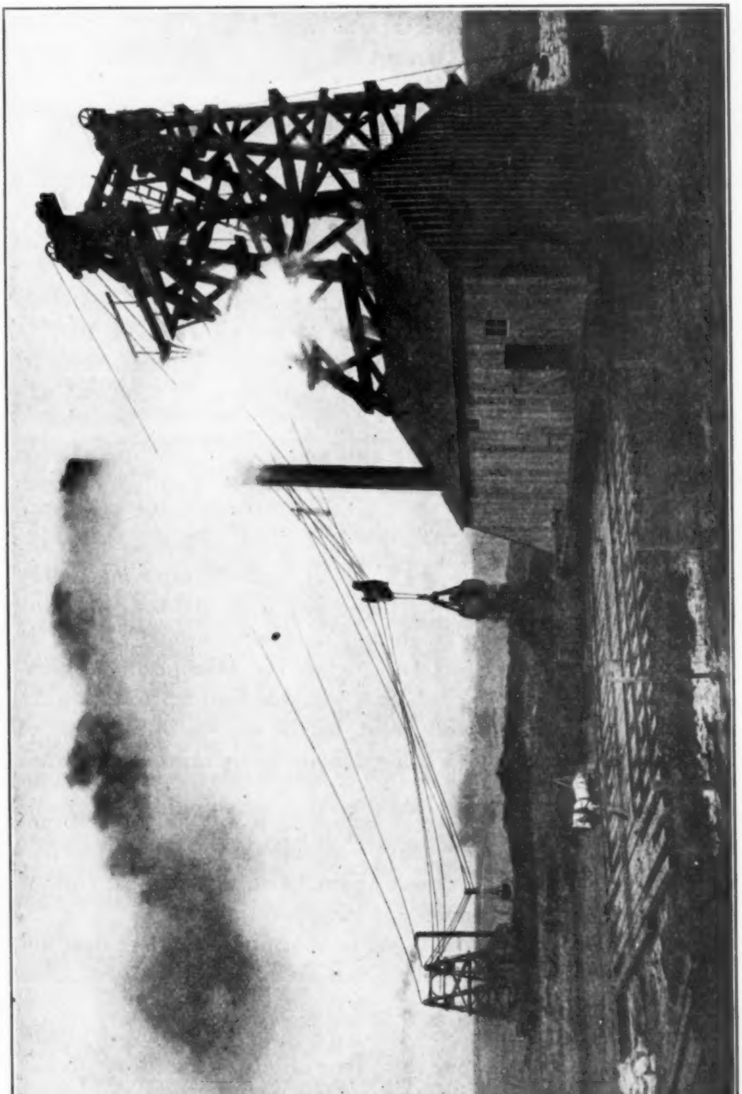


Fig. 1 Duplex Cableways

criticise in what he has said, but what I do criticise is that he has left so much unsaid.

3 Fig. 4 of his paper shows a type of ore-unloader that has proved to be the most efficient grab bucket five-ton unit machine in operation for taking ore from vessels.

4 Each machine is equipped with an electric man-trolley with a turn-table. The operator can with ease be making three or four distinct and different operations at one time, such as racking the trolley, turning the bucket and hoisting or lowering it. It is common practice for these machines to make a round trip in from 30 to 35 seconds.

5 The four machines at Conneaut unloaded the Steamship Hoover & Mason with 9202 gross tons of ore on June 14, 1907, in 7 hr. 12 min., which was an average of 326 tons per hour per machine, including cleaning up. The first hour each machine took out 429 tons, and the last hour 150 tons.

6 There were six of this same unit and type of machine started last season at Fairport, and before they had been running a month and a half they unloaded the steamship Cole with 11 385 gross tons of ore in 8 hr. 58 min., averaging 215 tons per machine per hour.

7 There are four machines in Cleveland, with the same machinery part equipment, but in addition equipped with 200-ton weighing bins, which can accurately weigh and deliver a carload of ore in about 7 sec. This seems remarkable when you take into consideration that the scales have to handle a variable load up to 250 tons. I call especial attention to this point, as the successful weighing of ore from vessels to cars is now being demanded by railroad and dock companies.

8 There are many other docks equipped with this type of unloaders; and the summary of what I wish to say in this connection is that they take ore out of vessels about twice as fast as any other type of five-ton unit machines now in operation.

9 Referring to Fig. 5, the bridges, as shown, are 565 feet over all, and are equipped with man trolleys, but do not have the turn-table feature.

10 The first bridge built operates a 7½ ton grab bucket, and will handle 350 tons of ore per hour from the end of the cantilever to the middle of the bridge. The second, or last bridge, operates a ten-ton grab bucket and will handle 500 tons per hour.

11 Referring to Fig. 6, under the heading of blast furnaces I do not recognize the bridge shown, nor do I understand why this bridge

is chosen when the majority of large blast furnace plants are equipped with bridges of the same make as the unloaders referred to above, to say nothing of the parabolic bins, larries, transfer cars and furnace hoists, which means that this machinery handles a large portion of the iron ore from the time it leaves the vessel until converted into pig iron.

12 Referring to the handling of coal, I note there are eight illustrations of anthracite-handling machinery, and only six of bituminous. In 1907 there were 375 000 000 tons of bituminous coal mined in this country, and only 67 000 000 tons of anthracite coal, which means less than one-fifth as much anthracite coal handled as bituminous.

13 There seems to be quite a change in sentiment among coal dock managers in the last three or four years, since four years ago the larger and more important concerns believed in small unit dock hoists, breaking bulk on face of dock, and delivering into stock pile by means of cars or trolleys. The dock people now seem to be in favor of larger units, and the bucket that goes into a vessel delivering into stock pile, thereby saving rehandling and breakage.

14 Further, sentiment seems to be growing in favor of man-trolley equipments, similar to what has become the standard for handling iron ore. In fact, coal is now being handled with man-trolley equipments at New Haven, Worcester, Springfield, Boston, Charleston and Superior, as well as at other places, and there is a bridge of this type now being built in New York, which will be referred to later.

15 Some of the advantages of the man-trolley are that the operator is always with his work, and long leads of whipping lines, sheave supports, etc., are eliminated. Fig. 12 and Fig. 13 refer to an anthracite coal-handling and storage plant at Superior, which is about one-quarter of the total problem on this dock, the other three-quarters being the handling of bituminous coal, which is not mentioned.

16 The bituminous dock is equipped with what is probably the most flexible coal-handling machinery of the intermittent type now in operation anywhere.

17 The equipment consists of four electric unloading towers on the face of the dock; a line of pockets immediately back of the same, with three electric transfer cars on top of the pockets and back of the towers; four three-span bridges at the rear of the pockets and extending the width of the dock, which is 500 ft., and a line of pockets on the rear of the dock running parallel with the pockets on the front.

18 The bridges are equipped with special man trolleys of two styles, one for stocking coal and one for taking coal out of stock with shovel buckets.

19 The fast plants will unload vessels at an average rate of about 150 tons per hour per machine, including cleaning up, and there are always one or two bridges available to fill all current orders from day to day for the different kinds of coal that are not being unloaded from the vessel.

20 Another plant of this unloading type, is the one in operation at the National Tube Company's Works at McKeesport, Pa., which is one of the fastest of which the writer has heard.

21 Another type is the Milwaukee Coke and Gas Company's two machines at Milwaukee. These are steam driven and operate a two-ton bucket. On June 9 and 10 of this year they unloaded the steamer Powell Stackhouse with 10 002 tons of coal, and averaged, without taking out any delays, 175½ tons per machine per hour, including the cleaning up of the vessel with the grab buckets.

22 These two machines were in operation just seven months last season, and one-third of this time there were no vessels at the dock. The machines were actually working four months and twenty-two days, and unloaded in that time 586 000 tons of coal, which is the best record I know of for any dock.

23 Referring to Fig. 21, this is one of the modern rope-driven trolley bridge plants operating a three-ton grab bucket, and I wish to call attention to the Pittsburg Coal Company's No. 6 Dock at Superior, erected four years ago, which handles only 1¼ ton buckets.

24 The general scheme of the two docks is similar, but the No. 6 Dock with only 1¼ ton buckets has a record of unloading a vessel at the rate of 108 tons per hour per machine, including cleaning up, delivering the coal at the rear end of the cantilever, which is 290 ft. from the apron, while the best record given for the machines shown in Fig. 21 is 135 tons per hour per machine with three-ton buckets. This No. 6 dock is steam driven, and has the same rope system as the Milwaukee dock which was referred to above.

25 While these two steam driven, rope system bridge equipments are successful for handling bituminous coal, there is, as stated above, a growing sentiment in favor of man trolley and larger unit grab buckets, which is very strikingly illustrated in the new bridge which is being built for the Astoria Light, Heat and Power Company, at Astoria, Long Island, which is a part of the Consolidated Gas Company of New York.

26 This bridge is 603 ft. over all, and is equipped with a man trolley, handling a grab bucket that will take between six and seven tons at a grab.

27 The moving load on the bridge is 40 tons. The speeds are:

	Feet per minute
Hoisting	225
Trolley travel	1200
Bridge travel	75

28 This bridge will be ready for operation the latter part of July, and will be the largest coal-handling bridge crane.

Mr. George B. Willcox Salt and chemicals are noted in Par. 75 of Mr. Baldwin's paper, as substances that can be satisfactorily handled with belts, but the kind of belts is not specified. In the salt machinery business our experience has been that belts are suitable only for certain kinds of service as salt carriers.

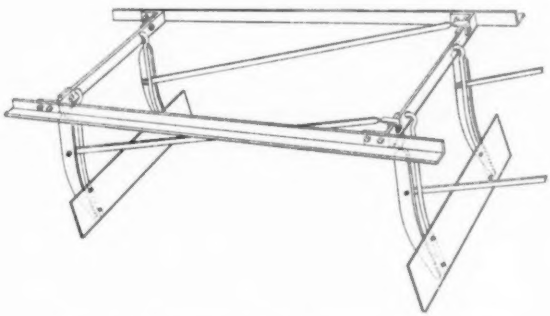


Fig. 1 Reciprocating Scraper Conveyor

2 For loading vessels in bulk, a belt conveyor located on the warehouse floor and delivering into a chute high over the wharf and discharging in the hold of the ship is satisfactory.

3 With 30 men wheeling salt in barrels and dumping at both sides of the belt, a 22-in. belt has delivered 150 tons per hour into the hold of a vessel, and with 25 men an average run is a cargo of 1100 tons in 10 hr. When navigation closes, the belt is cleaned and removed from the pulleys. For such service, namely, where a large quantity of salt is handled in a short time, probably nothing is so satisfactory as a belt. But where it is required to run

continuously, carrying only a small stream of salt, a belt is far from satisfactory. The state of the weather affects the operation. On a dry day newly made salt may discharge well from the belt surface, but a cold damp day causes it to stick to the belt, and the belt surface becomes wet and slimy.

4 Brushes and angle scrapers of many kinds, including wood and plate glass, have been used on the belt to overcome this difficulty,

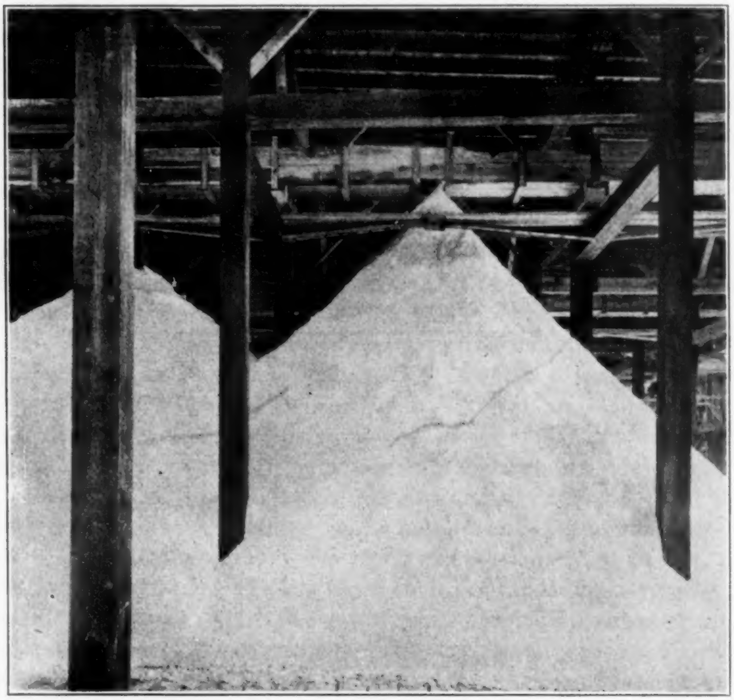


FIG. 2 CONVEYORS IN A SALT PLANT

but not successfully. A serious fault is that atmospheric dust accumulates on the belt surface, which travels through the air many miles a day, and in consequence rings of dirty salt accumulate around the rollers at the edges of the belt, finally dropping on the salt pile. Such dirt spots through the salt are very conspicuous and seriously affect the market value of the salt.

5 We have abandoned belts entirely for continuous service of the

class described, as for instance, removing the salt from a battery of grainers as fast as it is produced and distributing it in piles on the warehouse floor.

6 In such places the most satisfactory conveyor is of the reciprocating scraper variety, as shown in the accompanying illustration, Fig. 1. The side angles ride back and forth on slides made of 2 by ¼ angle about 8 in. long fixed to the side of the conveyor trough. The bottom of the trough is lined with ½ in. unfinished plate glass which can be had from any plate glass factory at about 25 cents per square foot.

7 These conveyors work back and forth very slowly. The slides do not wear rapidly and are easily replaced. Five conveyors of this variety have been in continuous operation night and day for nearly two years without repair and with no perceptible wear. They take the salt from 10 grainers, at the rate of 1000 bbl. per 24 hr., and deliver it in 20-ft. piles on the warehouse floor. Fig. 2 is from a photograph of one of them. One of these salt piles as shown is about 22 ft. high.

Mr. Charles Piez In giving his brief history of the development of the belt conveyor, Mr. Baldwin did not mention the introduction of what was known as the Robins patent belt some twelve years ago. This belt combined, according to the claims of the company exploiting it, "the advantage of a central thickened wear-resisting cover with stiffened edges, making the belt bend more easily at the center and better preserving its troughed shape between the sets of idler pulleys. This stiffening is done by running two or three plies of duck a part of the way in from the edges."

2 The purpose the inventor had in mind was evidently to give to a belt a largely increased carrying capacity by forcing it to assume the shape of a deep, continuous trough. To accomplish this, idlers of the form shown in Fig. 6 of Mr. Baldwin's paper were used, but with troughing pulleys arranged at angles fully 15 deg. steeper than shown in this cut. Large carrying capacities for narrow widths of belt, however, carried with them certain serious defects.

3 In order to secure flexibility the strength of the belt at the center was sacrificed by omitting some of the plies of duck. The large increase in load per unit area of belt required a heavy increase in the tractive force necessary to drive the belt and this was obtainable, owing to the limited width of the belt, only by largely increasing the initial tension. The resultant unit strain exerted on a belt that

was defective in structure quickly brought about a destruction of the bond between the fabric and the rubber coating and this destruction was effectually assisted by the lateral bending to which the troughing pulleys subjected the belt.

4 The remedy, of course, lay in materially reducing the depth of troughing, resulting in wider belts for the same carrying capacities, a much smaller degree of lateral flexure from the troughing rolls, lower unit stresses per unit width of belt and the advantage of using belts of uniform ply throughout their width. The progress of this development is shown in Mr. Baldwin's illustrations.

5 The original idler used for the patent or flexible belt was of the three-pulley type shown in Fig. 6 but with troughing rolls arranged at angles of about 40 deg. The next step was the reduction of the angle of the troughing rolls, some makers using 20 deg. and others 25 deg.

6 Fig. 5 of Mr. Baldwin's paper shows a concave roller developed by one of the users of a wide flexible belt for the purpose of overcoming the destruction of the belt due to the severe lateral flexure of the troughing rolls. The aim was to approximate in the idler the catenary curve that the belt would assume if supported only at the outer edges.

7 Fig. 7 of Mr. Baldwin's paper shows the unit troughing idlers arranged in substantially the same way, resulting in giving the belt practically a uniform curve in a transverse section. This form of idler has been adopted by a number of the leading manufacturers for the wider belt conveyors, and as is readily apparent, makes possible the use of a standard belt of uniform ply. This to my mind is a distinct advantage of the shallow trough belt conveyor.

8 The function of the belt in belt conveyors is twofold. First, it serves as the carrier of the material; second, it serves as the transmitting medium for the power necessary to move the material. The second function is undoubtedly best performed by a belt of uniform strength throughout its width; and if for purposes of carrying the material a form of troughing is adopted to which a uniform ply belt will readily accommodate itself, then there can be no question as to the best structure of belt to use for conveying purposes.

9 But while the shallow trough belt with idlers such as is shown in Fig. 6 and 7 has practically superseded the deeply troughed belt, there is at present a noticeable tendency to revert to the old type of flat belt. This tendency is illustrated by the increasing use for certain purposes of the idler shown in Mr. Peck's paper, Fig. 9. This idler is flat through the greater part of its length, the two

end sections being slightly convex to confine the material. In actual practice the end diameter is about $1\frac{1}{2}$ in. larger than the center diameter. The pulleys are very light, being made of pressed steel, and the shafts run in pivoted, chain-oiling bearings. The belts used with them are of uniform ply, usually protected with an extra thickness of rubber on the carrying side. For pulverulent, granular or sized material not exceeding $2\frac{1}{2}$ in. in diameter, and these classes, let me state, represent in volume the greater part of the material carried by belts, the slight troughing given by this idler is sufficient to retain the material on the belt.

10 The only charge that can be made against it, according to the standard set by Mr. Baldwin, is that the slip incidental to the difference in peripheral speeds of two portions of the idler is necessarily fatal to durability in the belt. But are the idlers of the construction shown in Fig. 6 and 7 of his paper free from slip? These idlers must be lubricated with grease because, as Mr. Baldwin very properly points out, oil would be hard to confine and contact with the rubber would cause rapid deterioration. But grease is if anything a retardant for very light loads and with the idlers spaced as closely as they are on the carrying side, the pressure per inch of journal bearing is small, particularly when the belt is running lightly loaded. Then, too, there is the friction due to the end pressure of the hubs of the inclined pulleys against the bearings. With such conditions existing, it is only natural to suppose that some slip will occur and the evidences of it are easily discernible to the practiced eye, for it is very rare to find all rollers on an installation running free when the belt is running light, and the idlers quickly become polished on the surface.

11 There is also the fact that a horizontal conveyor running light absorbs almost as much power as when fully loaded, the no load readings frequently being from 90 to 95 per cent of the full load readings. This slight difference can only be explained on the assumption of proportionately lower journal friction under full load conditions than under conditions of light loading. The inference is therefore that the idlers lag considerably under a light load and it is only fair to assume that they continue to lag to some extent under average load conditions. For after all, large capacity is secured with belts by carrying light loads at high speeds, and the average carrying loads are insufficient to operate grease lubricated journals at full efficiency.

12 Slip is therefore present in some degree in all idlers, and it is my opinion that with their greater lightness, their more perfect lubri-

cation, and their better alignment, the idlers in Fig. 9 of Mr. Peck's article are no more destructive to the under side of the belt than are those illustrated by Fig. 6 and 7 of the article under discussion. They are immeasurably kinder to the belt, however, in avoiding lateral flexure and on this account add very much to its durability.

13 It is almost an axiom in the belt conveyor art that belts wear out on the carrying side. Eliminating from consideration the deterioration caused by troughing, the reasons for the greater wear on the carrying side are:

a Impact of the delivered material;
b Action of pulleys of too small a diameter on the belt;
c Improper location and design of driving mechanism.

14 Mr. Baldwin has very fully covered the first cause but it may be interesting to add that where a mixture of fine and coarse material is handled, a very effective means of reducing the destructive action at the delivery point consists in inserting a short screen near the delivery end of the chute, thus permitting the fine stuff to sift through and form a cushion to receive the impact of the heavier pieces.

15 In regard to the second cause of the deterioration of belts altogether too little value is placed upon the use of pulleys of the proper diameter. I do not think it was Mr. Baldwin's intention to make it appear that the diameter of the driving pulley depends only on the width of the belt, yet that is the inference one unconsciously draws from Table 1. He does mention in another portion of his paper the minimum number of plies for various widths; but the relation between number of plies and diameter of driving pulley should be given in the table, for it is the number of plies and not the width of the belt that determines the size of the pulley. The action of a belt on a pulley is similar to that of a rope on a sheave and when the belt is forced around a pulley of too small a diameter, the outer layers of canvas stretch so materially with relation to the inner layers as to bring about a gradual destruction of the band between them.

16 While Mr. Haddock's experiments seem to indicate that there is only a slight gain in tractive effect when the pulleys exceed in diameter 5 in. for every ply, that is, that there is no great gain in driving power in a 30-in. pulley over a 20-in. pulley for a four-ply belt, yet these experiments touch only one side of the problem, and do not take into consideration the durability of the belt.

17 I feel most strongly that driving pulleys should have not less than 8 in. in diameter for every ply of canvas, and that no pulley

around which the belt passes under full load should be of smaller dimensions.

18 Take Fig. 10 of Mr. Baldwin's paper, for instance, the head pulley appears to be about two-thirds the diameter of the driving pulley. Assuming the belt to be 30 in. six-ply the layman would conclude from the table in Par. 41 that good practice would make the diameter of the driving pulley 30 in. and of the head pulley 20 in. Yet this combination would be a destructive one, for the belt is under as heavy a strain bending around the head pulley as it is on the driving pulley.

19 Now as to the third cause of deterioration, the improper location and design of the driving mechanism.

20 There is a fundamental principle which applies to all transmission media whether they be chain, rope or belt. It is this: make as few turns or bends under full stress as possible and avoid counter bends.

21 With this principle in mind there is only one proper position for the driving pulley and that is at the head or discharge end of the conveyor. In such a drive as is shown in Fig. 10 of Mr. Baldwin's paper, the compound drive, using pulleys of as small diameter as mentioned in Par. 38 and 39, might be mentioned as expedients but should hardly be cited as illustrative of sound practice. The conveying art, rapid and brilliant as its progress has been, is replete with examples of expediency, and it is now high time that consideration should be given to cost of maintenance and operation as well as to initial cost.

22 Let us now turn to Mr. Baldwin's formula for the horse power required for belt conveyors. The formula for horizontal conveyors

$$\text{h.p.} = \frac{C \times T \times L}{1000}$$

given by him suggests that the power is directly proportional to the load in tons per hour, yet that is not the case. For to get even approximate results with the formula the factor T should be a constant as well as C and Mr. Baldwin might have combined to advantage $C \times T$ into a single constant and given a table of values for it.

23 Some readings of belt installations are as follows:

a A 30 in. belt conveyor handling crushed bituminous coal. Conveyor is horizontal with 253 ft. centers. Speed 600 ft. per minute. Load carried at time of test 215 tons per hour. Conveyor is provided with self propelling

tripper. Readings, including motor and drive, are as follows:

	Horse power
Starting load empty	24.3
Running empty	11
Running loaded	11.7
Moving tripper	13.3

Belt empty

Mr. Baldwin's formula gives 9.18 h.p. without tripper.

Mr. Peck's table gives 11.70 h.p. without tripper.

b A 30-in. conveyor, 386 ft. centers, running horizontally at 348 ft. per min., carrying coke at the rate of 60 tons per hour.

The readings including motor and drive were as follows:

	Horse power
Empty	13.4
Loaded	13.4 to 14

Mr. Peck's table shows, with an allowance of 10 per cent per drive but without considering the motor efficiency, 11.36 h.p.

Mr. Baldwin's formula, substituting 60 tons for the capacity in tons per hour, gives 3.86 h.p.

24 Of course Mr. Baldwin states that the power should always be figured for the full capacity at the chosen speed, but what is the full capacity of coke on a 30-in. belt at a speed of 348 ft. per minute?

25 I am dwelling on this point to indicate that while the formula may yield results in the hands of a man who is privy to its peculiarities, it is not a good formula for general use.

Mr. T. A. Bennett With reference to Fig. 9 in Mr. Peck's paper and his mention of the four-roll idler which he considers best for modern practice, I would call attention to the so-called five-pulley idler shown in illustration herewith and mentioned in Fig. 7 of Mr. Baldwin's paper. This latest development of the idler form avoids the crease at the center of the belt where the greatest load comes. It conforms to the natural curve of the belt and retains the valuable feature of a center horizontal pulley. This horizontal pulley is necessary for guiding purposes, as without it the belt rides up and down the inclined idlers according to its loading.

2 With reference to belts Mr. Baldwin says truly in Par. 20 that "the belt conveyor industry has been built up mainly on the remarkable showing of the rubber belts made by Mr. Robins." I expected to have to call attention to the absence of the patent belt in Mr. Baldwin's paper, but I see this omission has been well brought out in the discussion.

3 It was found that the requirements of a conveyor belt were:

a Extra rubber cover at the center where the greatest wear comes;

b Pliability to fit the curve of the troughing idler;

c Heavy edges to stiffen and strengthen it.

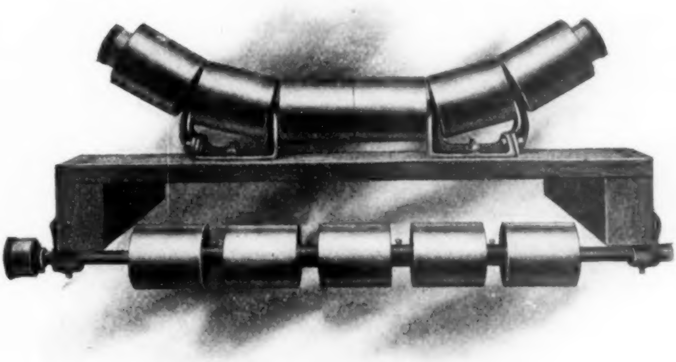


FIG. 1 FIVE PULLEY IDLER

4 I desire to correct the statement that the wear on the cover is uniform. We all know that the wear comes mostly in the center, the plies wearing away in steps rising to the edge, which is often not worn. These requirements were all met in practice by the patent belt made by the Robins Conveying Belt Company, a belt of uniform thickness, with a heavy rubber cover and extra plies of duck extending in part way from each edge, thus protecting and strengthening the edges and saving the use of unnecessary rubber. The resulting belt can be made as heavy at the center as will conform to the troughing idler and the heavy duck edges add stiffness and strength to the belt.

5 Regarding the waterproof qualities of rubber belts referred to in Par. 27, while it may be correct to state that they are not "abso-

lutely waterproof," they are practically so. The friction rubber is forced into the mesh of the duck and not only surrounds each thread but covers each ply, insulating it from its neighbor. An investigation of many belts after service has shown me that the moisture from a cut or puncture extends around it into the belt only about an inch; in fact we all know that the superiority of rubber belts in the resistance of moisture was the cause of their original adoption as driving belts in damp places.

6 Rubber belts with extra cover are the only ones that can stand the service mentioned in Par. 27 in handling dredgings, wet tailings, etc. On over fifty gold dredges in California, the Yukon and elsewhere, they are handling wet tailings; on the other hand, a 30-in. seven ply balata belt tried on a gold dredge in California lasted but 1330 hr. under exactly the same conditions under which the Robins Conveying Belt Company's belts preceding it had averaged 5000 hr. The question of moisture is not a determining factor in the life of a well-made rubber belt except at the very end of its service.

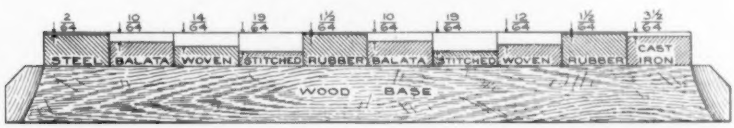


Fig. 2 Diagram Showing Relative Wear Under Sand Blast

7 The accompanying cuts show the relative ability of various belts and materials to resist abrasion. On a board are mounted the following materials, named in order from right to left: Cast iron, rubber belt, woven cotton belt, stitched canvas belt, balata belt, rubber belt, stitched canvas belt, woven cotton belt, balata belt, bar steel.

8 This block was subjected to the action of the sand blast by passing the nozzle uniformly back and forth over the center line for a period of 45 min. From this and many past experiments, we find the relative abrasive resisting qualities of these materials to be as follows, taking the volume of rubber belt worn away as being 1.0:

Rubber belt	1.0
Rolled steel bar	1.5
Cast iron	3.5
Balata belt, including gum cover	5.0
Woven cotton belt, high grade	6.5
Stitched duck, high grade	8.0
Woven cotton belt, low grade	9.0 to 15.0

9 Par. 30 of Mr. Baldwin's paper will bear correction regarding impairment in the strength and vitality of the fabric in rubber belts by the great heat of vulcanization. The "great heat" mentioned is only that due to a steam pressure of 40 lb. or about 290 deg. fahr., which, of course, is not enough to affect the cotton.

10 Par. 43 brings up the detail of the discharge from the belt. This discharge follows the path of a trajectory based on the velocity of the material and independent of the diameter of the pulley. The course of its fall is modified somewhat if the material clings to the belt: for clinging materials a clean discharge can sometimes be gotten only by high speed.

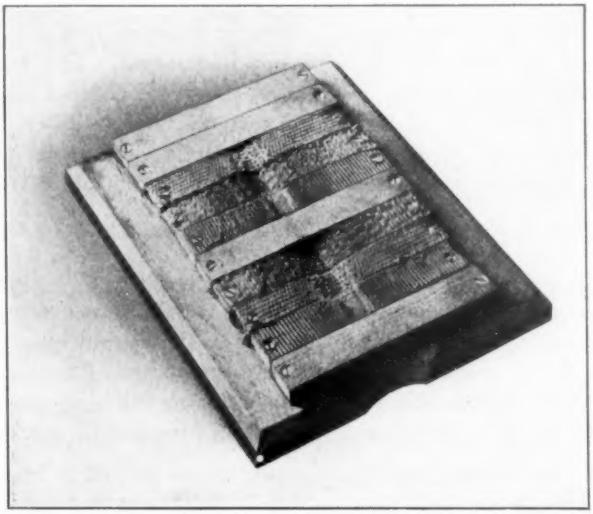


Fig. 3 Block Showing Results of Abrasive Tests on Different Materials

11 While there are mechanical difficulties to be overcome connected with the belts, drives, idlers, etc., the real engineering in the conveyor business is in the handling of the materials themselves. The success of an installation depends upon an intimate knowledge of these materials and of their chemical and adhesive action on belts, and most important, their flow in the chutes.

Mr. E. H. Messiter[1] Mr. Baldwin and Mr. Haddock have dealt with the subject of tension in conveyor belts. This is one of the most important factors in the design of conveyor drives.

2 Mr. Baldwin gives for the tension in the belt

$$\frac{\text{h.p.} \times 33\,000}{S \times W}$$

S is the belt speed and W the width of belt. This formula, however, gives the effective pull per inch, or the difference between the tension on the tight side of the pulley and that on the slack side.

3 The effective pull per inch is a figure that is useful in power transmission work, but in belt conveyor practice it is necessary to go further and find the actual tension in the belt. The reason for this is that there is such a variation in arc of contact and in the coefficient of friction of the pulley surfaces used in different drives that the relation between tension and effective pull varies widely.

4 To find the actual tension we resort to the old formula

$$\frac{T_1}{T_2} = 10^{0.000758 f \alpha}$$

which means that the logarithm of the ratio between the tensions on the tight and slack sides is equal to 0.000758 multiplied by the product of the coefficient of friction into the arc of contact.

5 Combining the above equation with the following formula for the effective pull, we are able to find the tension in any conveyor belt.

$$T_1 - T_2 = \frac{\text{h.p.} \times 33\,000}{S}$$

$$\text{Putting } \frac{T_1}{T_2} = 10^{0.000758 f \alpha} = R$$

we have for the maximum tension in a horizontal belt

$$T_1 = \frac{\text{h.p.} \times 33\,000 \times R}{S \times (R - 1)}$$

6 In regard to the belt conveyor with shallow trough which has been mentioned by Mr. Peck and Mr. Piez, a question suggests itself.

7 Remembering that the earliest conveyor belts were flat and

[1] E. H. Messiter, The Robins Belt Conveyor Co., Passaic, N. J.

that the only object of troughing was to gain an increase of capacity so that when a belt has to be renewed it will cost less money, it may be asked: unless we are going to get a substantial increase of capacity by means of a good deep trough, what is the use of troughing at all, and thereby sacrificing the simplicity of the single roller that suffices to carry the flat belt?

8 In connection with the subject of the diameter of conveyor belt pulleys, Mr. Haddock has proposed a minimum of 5 in. for each ply of belt while Mr. Piez contends that 8 in. should be the limit. The argument of the latter by analogy from transmission belts and driving ropes fails to take account of the wear on the face of the belt due to the impact of the material carried. In a properly constructed belt conveyor for anything but the lightest work the life of the belt is determined by the length of time its rubber cover withstands the abrasion of the material carried. We can therefore take some liberties in the way of smaller pulleys and counterbinding and can, I believe, use something not far from Mr. Haddock's recommendation for the minimum diameter of pulleys. Conveyor belts are certainly made today that wear out, as they should, by the abrasion of the rubber cover.

9 Passing to the question of the power consumption of belt conveyors, I agree with Mr. Piez that it would be desirable to have a uniform method of computation, but it will never be possible to test the constants for a conveyor of one make by trial with a conveyor of another design.

10 Mr. Baldwin has given a formula for horse power in which the constant is multiplied by the capacity and the length without direct reference to the belt speed. In its original form this formula has been in use for several years. It is as follows:

$$\text{h.p.} = \frac{1}{1000}\left[L\,(F_L\,T + F_s\,S) + H\,T\right]$$

in which

L = Length of conveyor.
F_L = Load factor.
F_s = Speed factor.
T = Capacity in tons per hour.
S = Belt speed.
H = Height through which load is lifted.

11 This formula is applicable to any belt conveyor and will show the difference in power consumption between a loaded conveyor and one running empty.

PROF. R. C. CARPENTER I merely want to say a few words on the subject of the paper of Mr. Tomlinson and on the subject of conveying machinery in the Portland cement industries. His paper gives the impression which I think to be an error, that belt conveyors are mostly used in these plants. He says that this industry more than any other depends upon conveying machinery for transporting raw material from where it is mined to the mill and through all the processes required to manufacture it and to the bins for the finished product. I think the history of conveying machinery in the Portland cement plants shows that it has been the source of more trouble and expense than almost anything else connected with the industry. The material to be handled is very gritty and heavy; consequently it is hard upon the conveying machinery. The cost of maintenance may be entirely out of proportion to the cost of installation if light conveyors are installed; for that reason it pays in the end to put in very heavy machines. I think Mr. Dodge will bear me out in this, for I know that he has been called upon very often to repair defects in the conveyor system of cement mills.

2 I understand from Mr. Tomlinson's paper that the screw conveyor is used but little in cement mills. My experience is very different from this; my observation indicates that it is largely used in cement mills. If well built it is very satisfactory and will stand up to its work better than any other type; its field however is limited.

3 It cannot handle coarse stuff satisfactorily. The material must be broken up to a quarter-inch diameter or less in order to be handled satisfactorily by a screw conveyor. From my experience in cement mills I should say the belt conveyor would handle satisfactorily materials not dusty or gritty; but a large portion of the material in the cement industry is very dry and dusty and it is extremely difficult to prevent the belt from scattering a great deal of dust, for which reason the belt conveyor has failed in many cases. I might say, however, that there are many belt conveyors in satisfactory use in the cement industry although the proportion to the total is small.

4 One form of conveyor has not been mentioned here, I think, one originally designed and patented by Mr. C. W. Hunt which has proved satisfactory in the cement industry, a car system handled by a continuous moving steel rope. This is one of the most satis-

factory systems of conveying that has been designed for the cement mills. If a car breaks down it does not stop the whole plant, as another can be supplied without stopping; whereas with other types if a part of one conveyor breaks down it may put out of service practically the whole plant.

5 Another word in regard to the elevator in cement mills. The material has to be handled and carried both horizontally and vertically, and so arranged that it can be moved over and from one level to another. Mr. Hunt introduced a system of automatic cars which may be pushed on elevators and carried up to different levels and shoved over to the desired point by trolleys or by hand. That system is coming into almost universal use and is as well thought of as any system at the present, in my opinion.

Mr. E. S. Fickes Mr. Peck in his very interesting paper does not describe a conveyor which has been found quite useful for handling finely divided dry materials and particularly those which are gritty and dusty. This conveyor consists of a tube with its axis either horizontal or slightly inclined; the tube is rotated on external bearings or trunnions and the material is carried forward either by the slope of the tube, if it is inclined, or by a helicoid or helicoidal arrangement of vanes or blades on the inside of the tube. The helicoid, or the vanes which replace it, revolves with the tube, thus carrying the material forward As the material is inside the tube it causes no wear excepting that due to its flow as it rolls spirally through the revolving tube from one end to the other. The bearings and trunnions are not exposed at all to the material being conveyed.

2 When the conveyors are small they can be built of rolled steel pipe, with rolled or cast helicoids forced or keyed into them so that there will be no motion between the helicoid and the outer shell. Larger conveyor shells must be built up of plates with either a continuous helicoid or suitable vanes fastened to the shell. Excepting in the case of very small conveyors, trunnion bearings are used, the conveyor being carried preferably by rolled steel tires turned true and rolling on the trunnion bearings. The conveyors can be driven either by gears, sprocket chains, or belts.

3 When the material is transferred from one length of conveyor to another, or when it is necessary to put an angle in the conveyor, the end of the conveyor is fitted into a box or transfer station into which the material is discharged and from which it is fed into the next section.

4 A simple arrangement to prevent the escape of dust from these transfer stations consists of a series of outstanding circumferential ribs on the shell of the conveyor which mesh into a corresponding series of stationary ribs on the transfer station with just enough space between to prevent them from rubbing. This arrangement is particularly useful if the materials are very gritty, in which case the grooves on the transfer station must be designed to discharge into it the dust which lodges in them. When the material is not gritty some simple form of packing ring or stuffing box can be used, although its use will increase somewhat the power required to run the conveyor.

5 The material is fed into the smaller conveyors by suitably shaped blades attached to the open end of the tube. When the conveyors are large enough a feeding spout is carried into the conveyor a short distance, and the end of the conveyor is covered by a head fitting the spout so as to prevent the material from rolling out of the conveyor.

6 This type of conveyor is particularly useful in chemical works and similar plants where lime, soda ash and similar materials are handled, the dust from which is often extremely disagreeable and annoying. It is also a very useful conveyor when such dusty materials have to be heated or cooled as well as carried from place to place in the works. If the material is extremely hot, it is necessary either to line a portion of the conveyor with fire brick or use water jackets to prevent the heat from burning or otherwise injuring the machinery. When the material is not too hot, a cast iron shell can be used with cast iron helicoids or blades for the parts exposed to the greatest heat, these being made so as to be easily replaced when destroyed. The cooling of the material as it is conveyed can be accelerated by spraying water on the outside shell of the conveyor, suitable guards being placed to keep the water off the bearings and from getting into the transfer stations.

7 Sometimes materials have to be gently heated or maintained at a uniform temperature as they are transferred from one department of the works to another, and this is accomplished by placing gas jets along the under side of the conveyors. If, however, more accurate regulation of the temperature is required and the material does not need to be heated too highly, a stationary steam heated pipe placed on the axis of the conveyor can be used, the shell being covered with insulating material to decrease the loss of heat by radiation.

8 It is evident that this type of conveyor is not adapted to

material which from dampness or other causes would tend to cling together and choke it, nor for the same reason can material containing large lumps be handled through the smaller sizes having a continuous helicoid which, with the shaft about which it is coiled, fills the entire tube.

9 The high first cost of this type of conveyor, as compared with its carrying capacity, greatly limits its field of usefulness, but for some materials and under certain conditions it is a valuable machine.

THE SPIRAL SPRING CONVEYOR BELT IDLER

Mr. E. G. Thomas A new form of troughing idler pulley for conveyor belts employs, instead of the usual cast iron pulleys, a flexible roller consisting of a spiral spring, 5 in. in diameter, rotatively supported in horizontally pivoted bearing boxes. The spring, under the weight of the belt and its load, stretches and sags to a smooth curve of approximately circular shape and the belt bends to a troughed form in contact with and supported by the spring across its entire width. As the spring is of the same diameter throughout, there is pure rolling contact at all points between the spring and belt. The spring has about two coils per inch of belt width.

2 The size of wire used for the spring varies according to the load to be carried and the commercial idler will be so made that the extreme fiber stress will be about 15 000 lb. per square inch. As

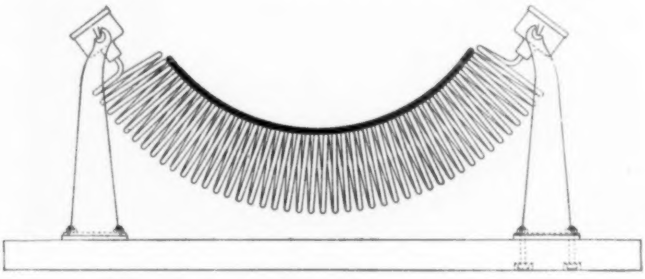


Fig. 1 Spiral Spring Roller for Belt

the variation in fiber stress as the spring turns is only from one-fifth to one-tenth of the maximum, a satisfactory life for the spring is assured. In all cases the spring is designed to have such strength that when the belt is carrying its maximum load the area of the cross section of the load will be the same as that of the ordinary 35

deg. three-pulley idler, so that the two forms have equal carrying capacity. Under any less load than the maximum the depression of the spring and the bending of the belt is less than the extreme.

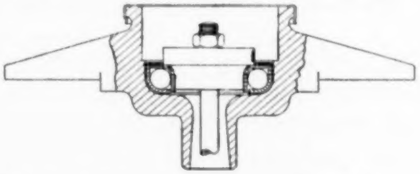


FIG. 2 BALL BEARING FOR END OF SPRING

3 The pivots of the bearing boxes permit them to accommodate themselves to the varying direction of the pull of the spring as it is depressed more or less by the changes in load, and as the strain on the bearing is chiefly that of thrust, ball bearings are used to advantage.

4 By the use of this idler, the curvature of the belt, instead of being concentrated along certain longitudinal lines, is equally distributed across its entire width and the degree of curvature is so small that wear from this source is practically eliminated. The

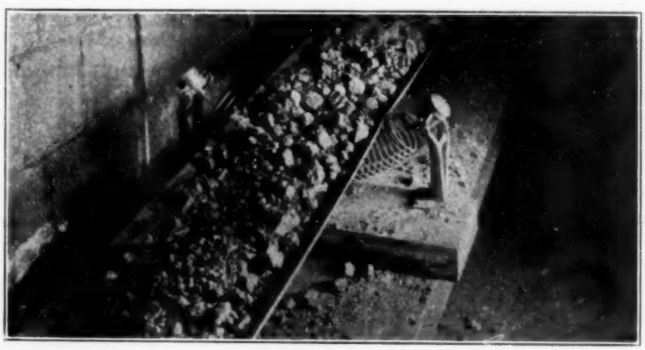


FIG. 3 LOADED CONVEYOR WITH SPIRAL SPRING IDLER

weight of the spring is much less than that of corresponding cast pulleys and the power required for its rotation in ball bearings is much smaller than is needed for the usual flat bearing with grease lubrication. The cost of lubrication will be reduced to the use of sufficient light oil at long intervals to keep the bearings free from

rust. In the conveying of material containing large masses, such as the tailings conveyors of dredges, the spiral spring idler has important advantages, and because of its elastic character, will receive no damage from the shock of a falling rock which may break the rim of a cast pulley.

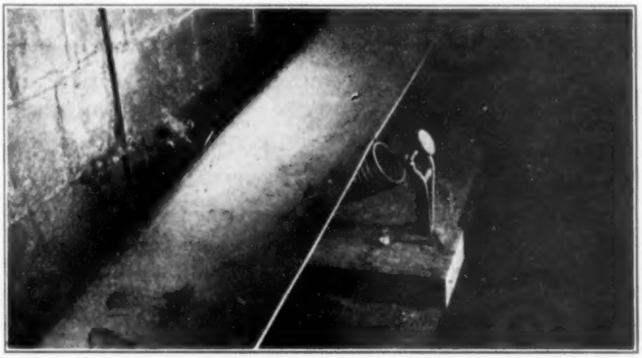


Fig. 4 Belt Running Empty

5 In the illustrations, Fig. 1 is an elevation of the idler under full load, Fig. 2 is a cross section of the bearing box, and Fig. 3 and Fig. 4 are photographs of the idler in use under a belt carrying coal.

Prof. H. Wade Hibbard offered a discussion from the point of view of the railroads. He said, in part, that it should be recognized that the railroads present quite a different problem from that of other industrial organizations. If anything goes wrong with the coal or other conveying installation of a private plant, the finances, or the reputation of the business, may suffer; but with a railroad engaged in public service, we must not forget that whatever happens the trains must operate; that those who wish to travel expect to go on time; and that shippers expect their goods to be delivered on time, and at the minimum cost of transportation. Therefore, when making an installation for coaling locomotives it must be remembered that the installation must not break down.

2 He disagreed with Mr. Baldwin as to the order of importance of the requirements for conveyors for railroad use. Freedom from shutdown is the first and most important point; that stands in a class by itself. Number 2 would be speed of coaling; 3, suitable location as

regards ground used and locomotive movement; 4, cost per ton of handling.

3 Incoming engines must be coaled, fires cleaned, inspected and oiled in the shortest possible time, and the engines quickly turned into service again. A terminal superintendent states that he has reduced the delay at the locomotive terminal from two hours to twenty minutes without the use of conveying machinery. Of course that is a timed record. Commonly, the men do not work at that speed. Labor saving machinery and appliances at overhead pockets, however, can be handled constantly at record speed, and therein lies one of the advantages of the conveyor, if ground area does not permit pockets with trestles.

4 The location of a coaling plant often presents a more fascinating problem than a game of chess, calling for high skill in operating, in civil engineering, and in motive power departments to avoid interference with main line operation. It necessitates making good use of the ground, whatever its area and topography, erecting suitable structures, preventing interference of locomotives with each other, and aiding in inspections and running repairs.

5 Par. 54 of Mr. Peck's paper refers to the cost of maintenance of the trestle type after the first 15 years. Some railroads make use of natural ground contours for the incline, thus avoiding the artificial structure.

6 Among the many items making up the total cost of coal, I would suggest that not too much attention be given to the cost of fuel to handle the coal. The vast amount of coal used for locomotives, its consequent cheapness, its haulage by the railroad for itself, make the amount used in the operation of the coaling station only a drop in the bucket. Too great attention to this item may divert attention from other more important items.

7 Finally, might I add that sufficient time ought always to be taken in the initial stage of the creation of this or any other engineering property, the stage of "contemplation," which properly precedes the stage of "design." Also, during both of these stages there should be maintained the closest coöperation among the departments o the railroad concerned, especially with the motive power department which is to use the property and will suffer most from its ill design.

Mr. J. McGeorge I had two questions to ask, one of which, upon the best method of conveying cement, has been answered by Professor Carpenter. The second question, which is closely allied with the

first, is that of the best method for getting material like cement into a conveyor. What is the angle at which finely ground cement will flow?

2 I cannot criticise Mr. Titcomb much because of omissions from his paper, for I know how difficult it is to get all relating to a subject in a single paper; but I did notice that he keeps his paper almost entirely to coal and ore handling machinery.

3 My firm is now building an installation of machinery for mixing concrete for some large government locks. We are putting up two crane bridges which take the sand and gravel from barges in the river by means of clam shell buckets, dumping the same into the concrete mixer, from the mixer dropping the concrete into a bucket, and laying it, all by means of those bridges, right in place.

4 There has been no trouble whatever in handling the gravel and sand by means of the grab bucket, etc., but there has been trouble in getting the cement in place to mix with the other ingredients of the concrete. We are handling cement in bulk, through a system of bins and inclined chutes. The bins are built in the form of hoppers and the chutes discharge into a locomotive bin car which runs an average of 600 ft. to the mixer, and the cement is then elevated into a tank and from that discharged into the mixer.

5 Our trouble is a little peculiar. The government engineers insist upon positive measurements. In these positive measurements we had trouble with the cement conveyors. One difficulty was to determine the angle at which cement would flow. I would be glad to hear from anyone who can answer that question.

Mr. William T. Donnelly This discussion has been extended to cover the art of conveying, broadly, but there still remains one branch which has not been touched upon, in spite of the fact that it deals with the oldest known system of conveying —the system that is handling the greatest quantities of material, and one that has been steadily at work since the beginning of the world.

2 I refer to the conveying of solid material by running water. The rivers and streams of the country are every year conveying countless millions of tons of material in their open channels, the force in use being the velocity of the water as it descends from higher to lower elevations.

3 At the present time, man is using the principle exemplified in this method of conveying in the dredging of harbors and rivers, and, to a limited extent, in the conveying of material for the construction

of dams, the velocity being imparted to the water by a centrifugal pump operated by a steam engine, thus bringing in purely mechanical apparatus, and bringing this system of conveying within the scope of the mechanical engineer.

4 It is rather a remarkable fact, that all such plants as have been used for this system of conveying up to the present time have been of a temporary nature; that is, they have been constructed and adapted to operate on a special contract for a limited time, after which they would be displaced or dismantled.

5 The economy or the low cost of handling material by this system is most remarkable. In dredging operations, the cost will range from 10 to 20 cents per cubic yard, with material conveyed a distance of from one-half to more than one mile.

6 From the necessity for conveying large quantities of material is developed the particular kind of conveyor best suited to meet the conditions, and it has seemed desirable to me at this time to call particular attention to this method of conveying solid material by water, as I believe that in the near future there will arise a demand for the continuous conveying of material by this system, and I am convinced that its lack of development has been due to the fact that, up to this time, there has not been a sufficient demand to call for the continuous conveying of material through pipe lines.

Mr. Henry Harrison Suplee This matter of conveying materials by hydraulics has been well carried out. One thing I think is worthy of mention, and that is the handling of ashes on shipboard. On nearly all the ocean liners the ashes from the boiler or from the furnaces are dumped in water and with a steam injector emptied over the side of the ship. The disposition of ashes was one of the most difficult problems on shipboard; it created a great deal of dirt and dust and labor. Now they are forced out with a stream of water and with perfect ease and freedom from dust. I think this device was the invention of one of our members, and is now used universally.

Prof. Wm. Kent A former president of the New York Steam Company, W. S. Andrews, now deceased, made some experiments, 10 or 15 years ago, on the carrying of anthracite coal in pipes by means of water pressure, and it was claimed that considerable success was reached.

Mr. A. B. Proal (presented by Mr. T. A. Bennett) I speak

for Mr. Proal, who is unable to be present and asked me to give some data regarding bridges and towers.

2 There are some unique features in connection with the bridge illustrated in Fig. 14 of Mr. Titcomb's paper. A 36-in. belt conveyor is made use of for handling coal into storage and a five-ton (coal) bucket for taking the material out of storage. This permits putting in coal or other material and taking it out at the same time, giving a bridge of double capacity. In coke plants where two or more kinds of coal are necessary, the field can be divided longitudinally and one kind of coal stored while another kind is being reclaimed. The capacity of the bridge is 700 tons of coal per hour into storage, but this can be increased by increasing the size of the belt. Coal has been taken out of storage by this bridge at the rate of 300 tons per hour. The storage field has a capacity of 750 000 tons and is arranged to be increased to 1 000 000 tons or more if desired. The bridge is equipped with a man-trolley, that is, the man rides with the bucket. Other features are, the load of the bridge on the further shear leg is carried by ball-and-socket bearings, and the bridge is driven through worm-gearing which removes the danger of the bridge running away in a wind storm.

3 The two towers illustrated in the back of Fig. 18 of Mr. Baldwin's paper have established averages, respectively, of 156 and 146 tons per hour in cleaning up 10 000 ton boats. It is important in speaking of tonnage to note that this refers to coal, not ore.

Closures to the Discussion by the Several Authors

Mr. G. E. Titcomb To Mr. Pattison, and others, the author would say that dealing with the subject from the standpoint of personal experience, he obviously had not the data essential to a general report, and could merely recognize the art practiced by others.

2 Mr. Miller's outline of the usage possible to cableways is accepted as authoritative, though our practice rarely calls for the employment of a device with long free spans in transmission of loads. Mr. Pattison directs our attention to the performance of a number of machines at various points, and on the whole his facts are indorsed by references of the author concerning the different types of machinery employed. A point must be made of the apparent discrepancy in noting the importance of anthracite coal-handling machinery: a natural consequence of the greater prominence resulting from the

early attention to the employment of mechanical means, and of the special type of successful apparatus inaugurated.

3 Our summary of the methods now in vogue for taking care of the immense tonnage of bituminous coal may be considered a sufficiently clear indication of the ability brought to bear in devising ways and means to handle it with expedition and economy. The available statistics will eventually serve as cumulative evidence of the progress following intelligent and systematic effort to perfect facilities.

Mr. S. B. Peck The subject of belt conveyors seems to have been very thoroughly handled. I regret that some of the numerous other successful types of conveyors have not elicited fuller discussion.

2 I agree with Mr. Piez, particularly as to the size of the pulleys and the location of the driving point, believing that the Transactions of this Society should show the best practice rather than what is occasionally expedient or permissible, and should even anticipate as far as possible the good practice of the future. The type of roller support shown in the lower half of Fig. 9, which troughs the belt but slightly and is supported by shafts runnning in pivoted chain-oiling bearings, has every advantage in its favor, though it requires a little wider belt and therefore greater first cost. I believe general experience shows that this objection disappears as soon as the real merit of a device is recognized, and I confidently expect to see this type of roller support very largely supersede the other forms in the next few years.

3 Mr. Messiter asks, following the same line of thought, why use a troughed belt at all: to which I would say that the slight troughing by the roller above referred to gives a practical increase in the carrying capacity of the belt much greater than might be supposed, the slight turning up of the edges checking the tendency of the material to spread laterally and dribble off the edge of the belt.

4 Mr. Bennett has criticised the four-roll idler where a deep trough is required, shown in Fig. 9, on the ground that a central horizontal idler is necessary to make the belt run true and prevent a crease in the center of the belt. My experience with a large number of the four roll installations has not shown that either of these objections exists, at least with a belt of uniform structure.

5 I believe this type of belt least liable to localized bending, and having the strains caused by tension uniformly distributed, it has the least tendency to the separation of the plies so fatal to the life of a belt.

6 An advantage is claimed for the belt referred to as the "Robins patent," in the added thickness of rubber cushion in the center, due to the omission of some of the plies. I had occasion recently to examine an installation consisting of a large number of conveyors with belts of this type, supported on three roll idlers, with sides inclined about 35 deg. All these belts, in service for some time, were nearly worn out on the carrying side above the inclined side rollers, due presumably to the impact of the material at this point, while there was still a cushion of rubber in the center $\frac{1}{8}$ in. thick or more, from which obviously no benefit would ever be received. The natural conclusion was that the distribution of this rubber over the entire belt would have prolonged its life.

7 Professor Carpenter brings out very clearly in his Par. 3 the limitations of the belt conveyor for cement work, also speaking of the extended use of the screw conveyor, and some of the limitations to which I had also referred. The type of conveyor, however, which seems to be largely superseding both of these in the cement mills, is the pivoted overlapping bucket carrier which I have shown in Fig. 16. Its somewhat greater initial cost has perhaps prevented its more general adoption earlier, but its advantages, judging from the large number installed in the most recent cement plants, seem to be now quite generally recognized. This conveyor is perhaps the most reliable, and least subject to derangement, and it is equally adapted to fine and dusty or coarsely broken and gritty materials as well as to materials at a very high temperature; and it combines in the same machine the ability to convey and elevate. Both Professor Carpenter and Mr. Tomlinson have advocated the use of cars in cement mills, and after all, this carrier amounts to an endless train of small cars. Mr. Tomlinson offers the criticism, in his Par. 18, that these carriers are excessively heavy for the load, which I think is hardly borne out by the facts. The following are the weights per running foot of some of the sizes in more common use, together with the amount of material handled per running foot.

Size Feet	Weight per ft. empty Pounds	Capacity per ft. material 90 lb. per cu. ft. Pounds
24 by 18	70	75
24 by 24	75	100
24 by 30	80	125
24 by 36	85	150
30 by 36	115	180

8 Comparison with the weight of ordinary industrial cars, or even freight cars, will show in favor of the conveyor, which, particularly in the larger sizes, weighs considerably less than the material handled. As to efficiency, by which I take it Mr. Tomlinson means the power necessary to operate, comparison with either type of car shows about the same advantage in favor of the conveyor.

9 I also want to take exception to the statement made by Mr. Tomlinson in Par. 15, that the pivoted bucket carrier cannot handle heated material. Carriers of this class are in very successful use in both cement and chemical works, handling both finely ground and coarsely broken material at nearly white heat. Mr. E. S. Fickes, who has contributed a very interesting discussion on the subject of conveyors at this meeting, I think will bear testimony to this statement from personal experience.

10 A point not brought out by my paper is the advantage of chains of long pitch—18 to 24 in. and upwards. These, by their fewer joints or articulations, reduce the weight and cost of the conveyors and their elongation from wear. The cost of lubrication and repairs is also correspondingly reduced, and usually chains of long pitch afford better opportunity for the secure attachment of slats, pans or buckets. Their extended use has been made possible by what is commonly designated as an equalizing gear. As the sprocket wheels are polygonal in form, with usually six to ten sides, there is considerable variation in the lengths of the controlling radii, varying the chain speed and causing a jerky motion. The pitch circle of the equalizing gear wheel has a varying diameter, or a number of lobes corresponding to the number of sides of the sprocket wheel, and the pinion is correspondingly eccentric. The regularly varying speed ratio therefore exactly counteracts the speed variations that the differing diameters of the sprocket wheel would cause. This device is the invention of Mr. James M. Dodge, a well-known member and Past President of this Society, to whose enthusiasm, fertility of mechanical suggestion and keen appreciation of commercial utility, the development of the conveying art is largely due.

Mr. C. J. Tomlinson I wish to correct the impression given by my paper, in regard to the relative proportion of use of belt conveyors and screw conveyors. As Professor Carpenter states, the screw conveyor is largely used in this work. It is a device that can be readily installed where the head room is restricted, and its large use is more the result of an effort to economize in the cost of plant erection than of the superior advantages of the device in operation.

2 If a belt conveyor is driven at comparatively slow speeds, about 200 to 250 ft. per min., much of its objectionable dust can be avoided. The additional cost of the wider belt required should not outweigh the advantages it has in operation over the less expensive screw and casing.

3 Mr. J. McGeorge asks the angle at which cement will flow. Cement will flow into a bin, and settle, at an angle of about 10 deg. from the horizontal; after it has settled and the air has escaped, it will stand in the bin at about the same angle from the vertical. This angle of repose is variable, as the cement falls in miniature land slides when being drawn, and in the process of one of these slides will deliver through a hose. I have measured the angle in a number of bins and found it varying from 64 to 60 deg. from the horizontal, and at the latter angle it is very stable. A trough from a grinding machine will operate at an angle of 34½ deg. with free delivery at all times, but cannot be depended upon to start itself if the flow is shut off at the lower end.

4 In a dusty location this material settles on beams or ledges very like a heavy snow, clinging together with an apparent angle of repose of 75 deg. This peculiarity has caused trouble in belt conveyor tripper boxes, because of the tendency of detached particles to lodge and cone up about the heavy stream, preventing its free discharge. A box giving good satisfaction with a material having an angle of repose of 45 deg. will often give trouble with cement.

5 The phrase "angle of repose" is somewhat misleading, as materials often have one angle of flow through a smooth spout, another at which they will flow over themselves, and another at which they will stand when being drawn from a thoroughly settled storage. These angles all affect the capacity and arrangement of conveyors, bins and storage yards.

6 In Mr. Peck's closure several points are raised, which seem to require a reply.

7 An 18 by 24 in. pivoted bucket conveyor, with a capacity of 75 lb. per ft. and operating at a speed of 50 ft. per min., will handle the raw material for a 5000 bbl. plant and the clinker for a 7000 bbl. plant. For this reason this form of conveyor in the larger sizes is little used in cement plant work. The mechanical efficiency of a simple type of car may be increased by increasing its cubical contents, but an arrangement that limits the use of these cars to a continuous chain also limits this increase by causing the capacity of the chain to become so great as not to be applicable to the average condition.

8 Large bins and intermittent service, or a very slow speed of travel and a correspondingly large conveyor, are possible recourses but serve to emphasize the limitations of this arrangement. The average pivoted bucket conveyor has a pair of carrying wheels located above the center of gravity of the bucket and within 6 in. of its surface. When handling a highly heated material the problem of retaining a lubricant in these journals is a difficult one.

9 The application of a grease box and pad, so successful in railway practice, is prevented by the fact that the entire car is revolved as it passes around the various sprockets. Again, their large number precludes the adoption of any but the simplest oiling devices, and also prevents more than a minimum of individual attention.

10 In cement plant work, these conveyors are in many cases being operated without any attempt at lubrication. Compression grease cups are used almost exclusively on conveyors in this industry. The heavy grease is a good lubricant, with the slow speeds, and is also not so liable to wash dust particles into and along the journal. Its movement is a slow one away from the cup and toward the open end or ends, where it forms a ring that very effectively prevents the dirt from entering.

Mr. C. Kemble Baldwin It is to be regretted that the discussion of this subject has been contributed only by men interested in the manufacture of conveying machinery. When the paper was published the writer sent copies to a dozen large users of belt conveyors asking for comments and such power and cost data as would be of general interest. While many of these users have data which would form the most valuable part of a discussion on the conveying of materials, giving the results of actual experience unbiased by commercial considerations it is extremely difficult to persuade a works manager to put his notes into shape for publication.

2 Mr. Willcox discusses the use of belt conveyors in salt manufacturing plants. He states that while satisfactory for handling salt from warehouse to vessels in large quantities, they are not satisfactory for handling small quantities from grainers to storage building because of the difficulty of cleaning the return belts.

3 The writer cannot agree with this, because he has designed and built within the last six years several very successful plants for handling salt direct from the open pans and vacuum pans. The salt from the open pans contains about 50 per cent hot water, and the main claim for the belt conveyor is, that the salt comes in contact only with

the rubber belt, which will not discolor it. There is no difficulty in cleaning the return belts by rotary brushes. After a careful investigation one of the largest salt companies in the country recently made two extensive installations of belt conveyors for handling grainer and vacuum salt. While the reciprocating conveyor described by Mr. Willcox works satisfactorily, it will be noticed that the metal scrapers work in the salt, and must in time drop pieces of rust. It is for this reason, that belt conveyors and not reciprocating conveyors were adopted by the company previously mentioned.

4 In service of this kind we find it best to paint all metal parts of the conveyor with white lead or Atcheson graphite, including the finished surfaces. Rubber belts are used throughout.

5 In the first four paragraphs of his discussion, Mr. Piez makes the following statements:

a That the purpose of the original troughing idlers used with the Robins conveying belt was to *force* the belt to assume the shape of a deep continuous trough.

b That to secure flexibility in the belt, strength was sacrificed.

c That owing to increased loads per unit area of belt due to the deep trough, it was necessary to increase the initial tension largely.

6 From the above he draws the conclusion that, the belt being "defective in structure," the increased "unit strains" brought about a destruction of the bond between the fabric and the rubber coating, which was assisted by the lateral bending due to troughing. The remedy he states was reducing the depth of trough, thus decreasing the load per foot of belt and the unit stress in the belt.

7 The Robins Patent Belt was made with a flexible middle, part of the plies being stopped off at a varying distance from the edges. This formed a belt that would trough of its own weight and had stiff edges which not only supported the trough between idlers, but gave the desired tensile strength. Being flexible, it was not necessary to "force" it to assume a deep trough, and having extra plies of duck at the edges its strength was not impaired in any way. That the belt is not inherently defective in design is shown by the fact that belts of this type are still running after ten years of continuous service.

8 In Par. 3 Mr. Piez states that the difference in "load per unit area" between a shallow and a deep trough causes such increased tension in the belt that it is destroyed thereby. In Par. 11 he

also states that "the no-load readings are frequently from 90 to 95 per cent of the full-load readings."

9 These statements are difficult to reconcile, because if there is only 5 to 10 per cent difference between no-load and full-load, there can hardly be a heavy increase in the tractive force necessary to drive the belt on account of the slightly greater load of a deep trough, over that of a shallow trough.

10 In Par. 5 to 9 inclusive Mr. Piez states:

a That the trough of the idler was reduced because the lateral bending destroyed the belts.

b That the idlers illustrated in Fig. 6 and Fig. 7 of the writer's paper are "shallow-troughed."

c That the tendency is toward the use of an idler that is flat the greater part of its length, the end diameter being 1½ in. larger than the center diameter.

11 The first troughing idlers built by Mr. Robins some 13 years ago were of the three-pulley type illustrated in Fig. 6. The inclined pulleys were about 40 deg. from the horizontal. The angle was soon changed to 25 deg. and 30 deg., depending on the width of the belt and the duty. Mr. Piez calls them "shallow belt conveyors" and evidently agrees with the writer that they are correct. The reduction of the angle was not due to belt troubles, but to the fact that it was soon discovered that there were practical limits to the depth of load that could be put upon a belt, as has been explained by the writer under Speed and Size of Belts.

12 As the deepest practicable load could be obtained by less trough, an angle was chosen that would give: *a*, a reasonable capacity for the width of the belt; *b*, sufficient trough to support the belt between idlers, so that the minimum number of idlers would be required; *c*, enough trough to center the load properly, thus insuring a straight running belt. The angles chosen, namely 25 deg. and 30 deg., have proved so satisfactory that they have been adopted by most manufacturers.

13 The writer was not aware that there is a tendency to revert to the old type of flat belt—as practically all of the belt conveyors being installed today use idlers with pulleys inclined at 25 deg. or 30 deg. A belt supported on the idler described by Mr. Piez (lower cut Fig. 9 of Mr. Peck's paper), which is flat the greater part of its length, the edges being turned up ¾ in., has all the disadvantage of a perfectly flat belt, as a trough of ¾ in. would not materially increase its capacity over

that of a flat belt. It would not in any way aid in supporting the load between idlers, thus requiring more idlers, and would have no influence on centering the load. Its place, therefore, is in the class of flat belt-conveyors which are practically of the past.

14 In Par. 10, 11 and 12, Mr. Piez states:

a That there is a slip between belt and idler pulleys of the type in Fig. 6 and Fig. 7, due to the retarding action of grease and hub friction.

b That the no-load readings are frequently 90 to 95 per cent of the full-load readings.

15 The writer, in Par. 15, calls attention to the fact that grease lubrication increases the power consumed, but gives good reasons why oil should not be used.

16 The claim that the idler pulleys quickly become polished on the surface, due to the slip of the belt, is not the case on the conveyors designed and built by the writer. It should be kept in mind that the average belt speed is from 200 to 400 ft. per minute. Therefore the pulley speed is so slow that the weight of the belt alone will turn the pulleys without difficulty. If there is any slip it is not apparent either in the polishing of the pulleys or in damage to either belt or idlers.

17 It is surprising what a small difference there is between the no-load and the full-load readings on the belt conveyors cited by Mr. Piez. It can be explained only by a lack of proper attention to the design and workmanship of the bearings of the idlers and the proportioning of the other parts of the conveyor and its speed to the work to be done—as the writer finds from the results of hundreds of carefully tabulated tests, that the no-load readings vary from 35 to 75 per cent of the full-load readings, depending on the length and speed of the belt and the amount of material handled.

18 The power required to operate a conveyor may be divided into that consumed by the driving mechanism and that consumed in turning over the idler pulleys. The power required by the former per foot of conveyor does not of course increase proportionately with the length, while the power necessary for turning the idler pulleys varies directly with the length and load of the conveyor. We find in consequence, for short conveyors where the greater part of the power is consumed by the drive, that the difference between no-load and full-load readings is frequently small; whereas for long conveyors, in which the power required by the driving mechanism is a less important factor, the difference may amount to 50 per cent or more.

19 In Par. 62 are given certain percentages to be added to the computed horsepowers required by short conveyors, on account of the predominance of the power consumed by the driving mechanism.

20 Concerning the question of conveyor drives, Mr. Piez says, in Par. 13 to Par. 21, inclusive:

a That the writer places too little value on the use of pulleys of proper diameter.

b That each pulley around which the belt passes under full load should have a diameter of not less than 8 in. for each ply of canvas.

c That the drive shown in Fig. 10 of the writer's paper would be a destructive one to the belt.

d That the drive of a belt conveyor should be located only at the head or discharge end.

e That the multiple pulley drives described by the writer are expedient and not based on sound practice.

21 In Par. 40 the writer cautions against the use of pulleys of small diameters. Table 1 gives "minimum" size of driving pulleys and was so given under the head of drives as a guide to engineers who are designing plants, to enable them to determine clearances.

22 The writer has found it impossible to give a hard-and-fast rule for the diameters of driving pulleys. As Tables 1 and 3 show, it has been found in actual practice that the minimum diameter should be from 4 to 5 in. per ply of canvas, depending entirely on the stress in the belt and on local design conditions. Mr. Piez disregards the fact that Table 1 gives "*minimum* sizes of driving pulleys." Larger pulleys are naturally used when possible.

23 No part of the conveyor design has been given more attention than the drives and a careful tabulation of hundreds of installations shows that the belts do not fail through separation of the plies due to pulleys of these sizes.

24 Mr. Piez's theory that the pulleys should have a diameter of 8 in. for each ply makes them unnecessarily large, causes very unsatisfactory chutes and, as pointed out in Par. 39, complicates the reduction from motor or engine.

25 Referring to the drive shown in Fig. 10, which Mr. Piez claims would destroy the belt, attention is called to the fact that while the driving pulley transmits motion to the belt, the one at the head merely acts as a bend pulley. The head pulley in such a drive may be smaller than the drive pulley.

26 The drive shown in this photograph has a 48 in. driving pulley, a 30-in. head pulley, with a 9-ply 36-in belt. The best proof that this drive is *not* a destructive one is the record of this conveyor, which has been in operation over three years and has handled over 1,500,000 tons of mine-run coal. The original belt is still in use, and to use the words of the manager of this plant, "It looks good for another million and a half tons."

27 Mr. Piez, whose business has been mainly the manufacture of chain conveyors, would locate the drive of the belt conveyor only at the head end, as would be the case with a chain conveyor. The writer has for the past five years been using the multiple-pulley drive located anywhere in the length of the conveyor, and the results have been most satisfactory. Although hundreds of them are in use, there has not been a single case where the belt separated in the plies, due to the action of the drive. In this connection, accompanying tables will be of interest, both as to the location and the style of drives in three conveying plants designed by the writer.

28 Plant *A* handles mine-run coal on the 36-in. conveyors, capacity 500 to 700 tons per hour. 30-in. conveyors handle fine coal 300 tons per hour. The conveyors have been in operation over three years and have handled over 1,500,000 tons. The original belts are still in use. Fig. 10 is a photograph of drive of Conveyor No. 2. Since these conveyors were started three years ago, every chain conveyor in the plant, except one, has been replaced by a belt conveyor.

29 Plant *B* handles mine-run and fine coal, capacity 500 to 700 tons per hour. Conveyors have been operated about three years and have handled about 1 000 000 tons.

30 Plant *C* has just been started. After making careful inspections of Plants *A* and *B* and investigating all other methods of handling coal, the owners contracted for this plant.

TABLE I LOCATION AND STYLE OF DRIVES. PLANT A

No.	Width Inches	Length Feet	Inclination Degrees	Location of Drive	Type of Drive
1	36	600		Head End	Multiple Pulley
2	36	300		" "	Like Fig. 10
3	36	500	Level, and 20	Center	Multiple Pulley*
4	36	50	20	Tail End	Single " †
5	36	500		" "	Multiple Pulley
6	30	200	20	Center	" "
7	30	400	20, then Level	Head End	Single "
8	30	100		Tail "	" "

* Lifts Material 90 ft.

† Driven from head of No. 3.

TABLE 2 LOCATION AND STYLE OF DRIVES, PLANT B

No.	Width Inches	Length Feet	Inclination Degrees	Location of Drive	Type of Drive
1	36	300		Head End	Multiple Pulley
2	36	775		Head "	" "
3	36	411		Head "	" "
4	36	473	20, and Level	Center	" "
5	36	673	20, and Level	Head End	" "
6	36	661		Near One End Reversible	" "
7	36	82	20	Head End	Single "
8	36	196	20	Tail "	Multiple "
9	40	100		Head "	Single "
10	36	300	20	Tail "	Multiple "
11	28	177	20	Head "	Single "
12	36	120	20	Center	Multiple "

TABLE 3 LOCATION AND STYLE OF DRIVES, PLANT C

No.	Width Inches	Length Feet	Inclination Degrees	Location of Drive	Type of Drive
1	36	75		Head End	Single Pulley
2	36	315	20	Center	Multiple "
3	36	30	20	Tail End	Single " *
4	30	32		Center	" " †
5	30	50		Head End	" "
6	30	50		Head "	" "
7	30	400	20	Center	Multiple "
8	30	400	20	"	" "
9	36	640	20, and Level	"	" "
10	36	55	20	Head End	Single "
11	36	55	20	Tail "	" "
12	36	480		Center	Multiple " †
13	36	180	20, and Level	Tail End	" "

* Driven from Head of No. 2.
† Reversible Conveyor.

31 The writer believes that the foregoing is the best proof that the drive may be located at any point in the length of the conveyor, and that the multiple pulley drive, far from being an expedient, is based on sound practice and good engineering.

32 In Par. 22 to Par. 25 Mr. Piez discusses the writer's power formula and gives two examples. The results obtained from the power data presented by the writer apply to the belt conveyors manufactured by the Robins New Conveyor Company and the claim is not made that these figures will hold good on conveyors of different design.

33 In Par. 22 the writer laid stress on two fundamental points of economical conveyor design:

a The belt should run no faster than is necessary to carry the desired load. The speed may be determined from Table 2.

b The power required should always be figured for the maximum capacity at the chosen speed.

34 If these points are followed there will be a fixed relation between load and speed: therefore T in the writer's formula (load in tons per hour) is a function of both load and speed. While C is a constant for each width of conveyor, T is a variable depending on the desired load, so that if these were combined as suggested by Mr. Piez a large range of values for T would have to be used, resulting in a cumbersome table of doubtful value.

35 Mr. Peck's table of horse power, to which Mr. Piez refers, and the examples given by him in Par. 23, all indicate that they have left out of consideration the relation between speed and load.

36 Take example *A*, 30-in. conveyor handling coal, 253 ft. centers, capacity 215 tons per hour, speed 600 ft. per minute. Assuming coal to weigh 50 lb. per cubic foot, we find on referring to Table 2 that a 30-in. conveyor at 100 ft. will carry 145 tons of material weighing 100 lb. per cubic foot or 72½ tons of coal; to carry 215 tons therefore the speed should be about 300 ft. per minute instead of 600 ft. Naturally the lower speed will require less power and produce less wear and tear on machinery and belt. According to the writer's Formula 9, 18 h.p. will be required, but this is based on a belt-speed of 300 ft. per minute, and not 600 ft., which is bad practice.

37 Take his example *B*, a 30-in conveyor, 386 ft. center to center, speed 348 ft. per minute, carrying 60 tons of coke per hour. When handling such friable material as coke, the belt speed should be as low as possible to avoid breakage. The speed also should be no greater than is necessary to carry the load.

38 Taking coke at 25 lb. per cubic foot, 60 tons per hour, the capacity desired is about 4800 cu. ft. From Table 2 we find that a 30-in. conveyor has a capacity of 2900 cu. ft. at 100 ft. per minute, therefore the proper speed is about 165 ft. per minute instead of 348 ft. According to the writer's formula, this gives 4.25 h.p., 10 per cent being added for gears, and this would drive the conveyor as built by the writer. The excessive powers shown by Mr. Piez's tests are due in part to the excessive speed.

39 Mr. Messiter presents in Par. 10 of his discussion a formula in which a load factor is multiplied by the load in tons per hour and a speed factor by the speed in feet per minute. With this formula it is possible to compute the power required using a high speed and small

load. Should such a conveyor at any time receive the full load at the high speed it would result in failure of belt or drive. In the writer's formula, in which the speed and load factors are combined, and where T is a function of both load and speed, this possible error is avoided.

40 Mr. Bennett, in Par. 5 of his discussion, takes exceptions to the writer's statement regarding the waterproof qualities of rubber belts. The rubber friction is applied to the duck in a plastic state. As its consistency is that of putty, it can do no more than coat the duck without reaching its inner fibers. Therefore when affected by a large quantity of water the belts are destroyed, as described by the writer in Par. 27. On the gold dredges the service is extremely severe and the belts must be built to stand excessive abrasion due to large quantities of wet rock. Therefore it is a duty which requires not only waterproof qualities, but also abrasion-resisting properties. The uncovered Balata Belt mentioned by Mr. Bennett should never have been used in this place as it was not provided with any cover to stand the wear. It should have had a heavy rubber cover.

41 Mr. Bennett states in Par. 9 that the heat of vulcanization does not affect the cotton duck of rubber belts. Actual tests, however, show that there is a loss of from 10 to 15 per cent in tensile strength due to the heat of vulcanization.

42 Mr. Messiter, in Par. 2 of his discussion, takes exceptions to the writer's formula for tension in the belt. As he states, the formula gives only the driving tension and neglects the initial tension, due to the take-ups. The substitution he proposes, while correct, is too cumbersome, and that given by the writer gives values that seem close enough for practical purposes.

43 In this connection, the writer would call attention to his reason for presenting his paper, stated in Par. 74, "To give the engineers who were designing plants some data which will be of service in the preparation of preliminary designs." This purpose made it seem advisable to simplify and make as practical as possible all of the data presented, and to omit all unnecessary detail.

44 As this discussion has brought out some differences of opinion regarding principles and methods employed, it might be considered pertinent to add a word relative to the connection of the writer with the development of the belt conveyor industry, that members of the Society may better judge of the value of the data presented.

45 In the early days of Mr. Robins' activity he had associated with him James Barnes Humphrey, a young engineer of exceptional bril-

liancy, who personally designed, with no precedent to guide him, not only the various parts of the belt conveyor (many of which remain unchanged today), but also several large and difficult conveyor plants. These installations, although unique and daring in design, were so successful that the future of the belt conveyor as an industrial factor was immediately assured.

46 By his withdrawal from the work in 1900, followed shortly thereafter by his death, the industry can be said to have suffered the loss of a real genius. The writer, who was associated with Mr. Humphrey, and who carried on his pioneer work, desires that full credit be given the mind that rendered such invaluable services in the evolution of the belt conveyor.

47 The data presented represent the results of the writer's observation of thousands of conveyors, as Chief Engineer of a company manufacturing practically only belt conveyors, and all statistics and figures have been proved by actual practice.

No. 1196

THE THERMAL PROPERTIES OF SUPERHEATED STEAM

By Prof. R. C. H. Heck, South Bethlehem, Pa.
Member of the Society

The purpose of this paper is to compare, discuss and combine the best available data in regard to the specific heat of superheated steam, to determine as nearly as may be the true value and manner of variation of that quantity, and to derive a numerical table which shall be as reliable and convenient for general use as is the ordinary table of the properties of saturated steam.

2 The discussion will be based almost altogether upon the experimental researches of Knoblauch and Jakob and of Thomas. The former were published in the *Zeitschrift des Vereins deutscher Ingenieure,* in January 1907, at pp. 81 and 124; the experiments were made in the laboratory of technical physics at the Royal Technical High School, Munich, in the latter part of the year 1905; and the results were brought before this Society by Professor Greene at the Indianapolis meeting, in 1907, published in Transactions, Vol. 29. The paper of Professor Thomas, setting forth experiments made at Sibley College, Cornell University, was presented to the Society at the Annual meeting, 1907, and is published in Transactions, Vol. 29.

3 In each paper is given a set of curves for the specific heat at constant pressure, c_p, the individual curve showing how c_p varies with the temperature during the operation of superheating at some particular pressure. To superimpose the two groups of curves on one plane would be very confusing; they can be much more effectively compared by means of the "solid" or three dimensioned diagram drawn in Fig. 1.

4 As indicated, one abscissa of the base plane is the temperature t in degrees fahrenheit, parallel to the axis OT, the other is the absolute pressure p in pounds per square inch, parallel to OP, while c_p is the

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers

vertical ordinate. The primary curves, each on its own TC plane, are redrawn directly from Fig. 6 of Thomas' paper and from Fig. 11 of Knoblauch and Jakob. The latter figure is reproduced as Fig. 1 in

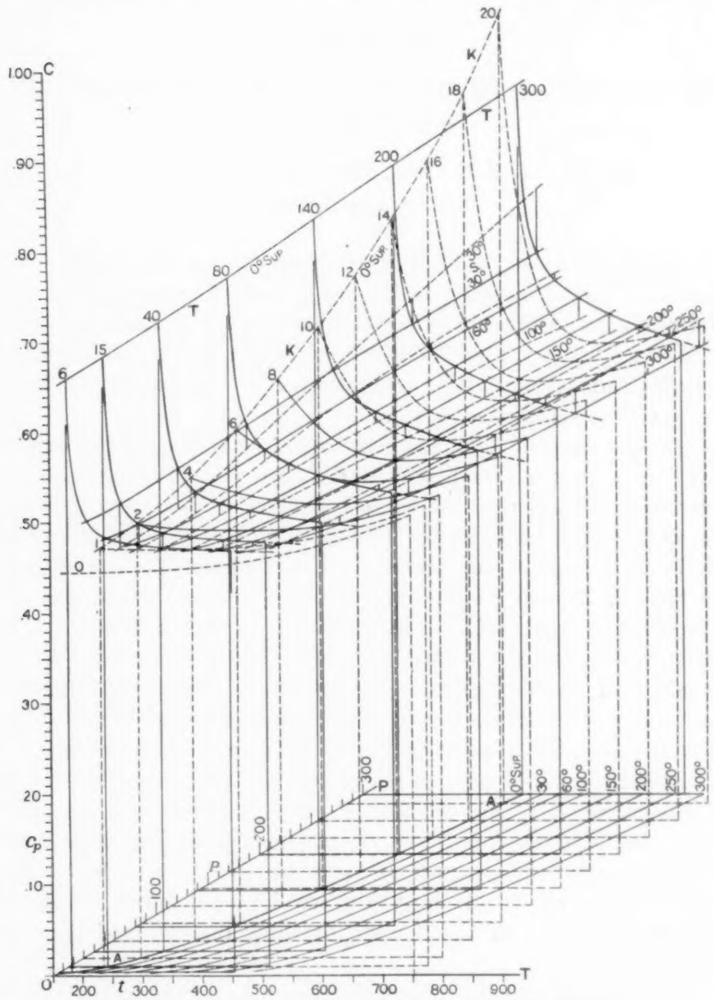


Fig. 1. Comparison of Curves of Specific Heat

Greene's paper. The law of variation of c_p with temperature and pressure, as determined by each experimenter, is now represented by a curved surface; and by drawing cross curves we have an easy means

of interpolating so as to get *T C* curves in both systems at the same pressures (see Fig. 4), or can use the cross curves directly for comparison.

5 The line joining the high initial points of the *TC* curves, at the left, is the line of saturation or of zero superheat; it lies on a vertical curved surface which cuts the base plane in the curve *AA*, the latter being the fundamental curve of relation between the temperature and the pressure of saturated steam. Along the saturation lines the letters *K* and *T* designate the respective results of Knoblauch and Jakob and of Thomas, and these letters will be used as abbreviations hereafter. The numbers on the *T* curves show pressures in pounds, while those on the *K* curves show kilograms per square centimeter, the relation being, 1 kg. per square centimeter = 14.22 lb. per square inch. The *T* curves are all in full line, except the dotted extensions beyond the limit of actual determination at 270 deg. of superheat. In general, the *K* curves are dotted, except the four which show actual experiment, namely, those for 2, 4, 6 and 8 kg. These curves are stopped at 400 deg. cent., or at 752 deg. fahr., although the determinations really extended only to about 700 deg. fahr., as is made clear on Fig. 4.

6 The cross curves are drawn for constant superheat. On the base plane *P T* a number of curves are put in at a constant distance, in the temperature direction, from the original pressure-temperature curve *AA*—or, in effect, this curve is shifted various distances to the right. From the intersections with the base lines of the *T C* curves, vertical lines are drawn to the curves themselves; or, more simply, measurements of 30 deg., 60 deg., 100 deg., etc., are made directly from the lefthand limiting ordinates to locate points on the curves. Through these points the cross curves are traced, and marked with the degree of superheat; and further, short verticals are extended from every intersection point on the *T* surface to the corresponding cross curve on the *K* surface, in order to show how far apart the two surfaces are, or to get points on a primary *K* curve at the same pressure as each *T* curve.

7 Examining Fig. 1, and bearing in mind the fact that the *K* curves beyond 8 kg. are got by extrapolation, we note that from 50 or 60 deg. of superheat out to the limit of the experiments the two sets of results agree very well; but that there is a marked difference near saturation and the beginning of a wide divergence toward the region of higher superheat. We shall now take up the two questions suggested by these discrepancies.

SPECIFIC HEAT AT AND NEAR SATURATION

8 Near to the boundary line which separates the superheated from the saturated state, it appears that there are some residual molecular attractions to be overcome, and for this reason the specific heat is higher at and near this lower limit of superheat than it is farther out. A number of considerations affecting the conditions just at this limit are set forth graphically on Fig. 2, where the base line is temperature fahrenheit. The four curves marked T, K, H_1

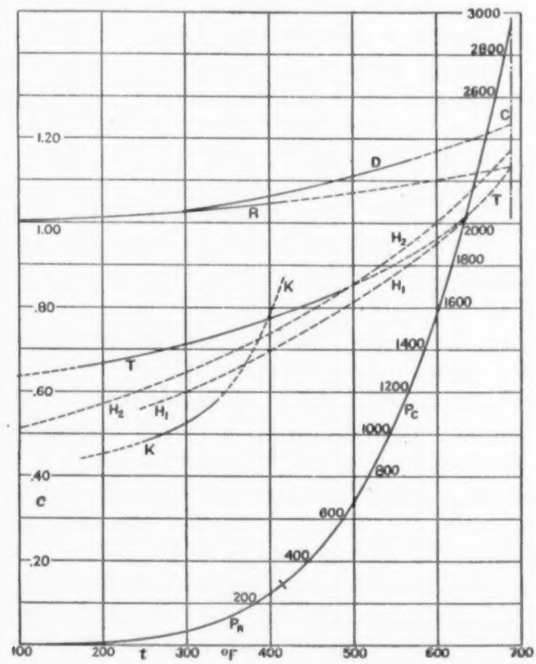


FIG. 2 CONDITIONS AT THE SATURATION LIMITS

and H_2 show the specific heat c_p of saturated steam along the saturation line, under various conditions or assumptions; the curves R and D give the specific heat of water, which rises well above unity with increasing temperature; the curve designated by the letter P shows the relation between pressure and temperature for saturated steam. The lower part of the last curve, marked P_R and extending to the cross-line at about 300 lb. pressure, gives Regnault's determination; the upper part P_C is from the experiments of Cailletet and Colardeau,

published in *Annales de Chemie et de Physique*, sixth series, vol. 25, 1892, p. 527. It runs up to the critical temperature as determined by these investigators, which is at 689 deg. fahr. and at a pressure of 2946 lb. per square inch. The numbers marked along this curve give the scale of pressure.

9 Of the two curves for the specific heat of water, the one marked R is from Regnault's formula,

$$c = 1.0000 + 0.000\,00222\ (t - 32) = 0.000\,000\,2778\ (t - 32)^2,$$

which is based on experiments extending to about 425 deg. fahr. The curve D is from Dieterici's experiments, published in the *Zeitschrift des Vereins deutscher Ingenieure*, March 1905, p. 362, as represented by the formula:

$$c = 0.9983 - 0.000\,0576\,(t - 32) + 0.000\,000\,64\,(t - 32)^2$$

Here the limit of actual determination was 300 deg. cent. or 572 deg. fahr., and this formula is to be accepted for the higher range of temperature.

10 The reason for presenting these apparently irrelevant data will now be made apparent. The specific heat curves K and T, for c_p at zero superheat, laid out with ordinates transferred to Fig. 4 directly from Fig. 1, and with the range of experimental determination shown in full line for each case, represent radically different ideas as to the manner of variation of this quantity. Knoblauch and Jakob derive a formula for the curve which passes through the inner ends of their four $C\ T$ curves, putting this formula into such a shape that it makes c_p equal infinity at the critical temperature of 365 deg. cent. or 689 deg. fahr., and remarking casually that the mechanical theory of heat calls for this value at that point.[1] The curve taken from Thomas' presentation conforms very well to the idea that since hot water and superheated steam merge into a common state at or above the critical temperature, the two specific heats, coming up along and just outside of the inner and outer limits of the condition of wet or saturated steam, ought to run into the same value, decidedly a finite number, somewhere near the critical point. That the latter is, in a general way, the correct view is the opinion of the writer, who further believes that the idea of infinite c_p at the critical temperature is based on a misconception.

[1] Zeuner, in his Technical Thermodynamics, at the end of art. 38, p. 312 of vol. 2 of the recent translation by Professor Klein, makes the same casual statement. The writer does not know who first developed the idea.

THE CRITICAL STATE OF WATER

11 In order to clear up the difference of opinion or of conception suggested in the last paragraph, and to get a reasonable indication of the trend of the initial c_p toward the higher ranges of pressure, we must consider more fully what is involved in the term "critical state" or "critical point" of a fluid. In one definition, it is a limiting temperature, above which the substance cannot exist in the liquid state; according to another, it is a condition where the two states of liquid and of superheated vapor merge into each other, the ordinary operation of evaporation into saturated vapor disappearing. Under the latter idea, the latent heat of vaporization, which decreases with the temperature of steam formation, will become zero at the critical point. Not to cite other writers, we may refer to the paper by Mr. C. V. Kerr on "The Potential Efficiency of Prime Movers," presented to the Society in 1904, and published in the Transactions, vol. 25, p. 920. It is there stated that by producing the two limit curves on the entropy-temperature diagram until they met, the point where the latent heat vanishes was found to be at a height of 975 deg. fahr. This method is only the roughest kind of an approximation; but when we compare the result with the value 689 deg. quoted above, there appears to be a wide discrepancy.

12 The experiments of Cailletet and Colardeau consisted in observing the simultaneous pressure and temperature of a small mass of water in a closed metal tube. They found that the single, definite relation between p and t, characteristic of all ordinary ranges, ceases to exist at 365 deg. cent., or 689 deg. fahr. We shall now proceed to show that this "critical point" is by no means the same as that defined above.

13 In Fig. 3 is drawn the usual pressure-volume diagram for one pound of water substance, but with a very small scale of pressures and a very large scale of volumes: *A B* is the line of water volume, or the inner limit of saturation, between water and wet steam; *C D* is the dry steam curve or the outer limit of saturation, separating wet steam from superheated. The horizontal line *B C* is drawn at the pressure corresponding to 689 deg. fahr., or at 2946 lb. The limit curves *A B* and *D C* are laid out according to determinations made at comparatively low pressures, and are therefore merely tentative; but they decidedly do not meet at the height *B C*. We must believe rather that *B C*, as determined by coming up from below, is one limit of a quite widely extended "critical state," of which the upper limit is at some

point like E which is located by the convergence of curves produced from $A\ B$ and $D\ C$.

14 The most distinctive characteristic of the ordinary mixture of steam and water, thermodynamically considered, is found in the fact that the operation of evaporation at constant pressure takes place isothermally. For a so-called perfect gas the isothermal operation follows the law $p\,v$ = const., and this is very nearly realized with any actual dry gas. The complete isothermal of steam, at any moder-

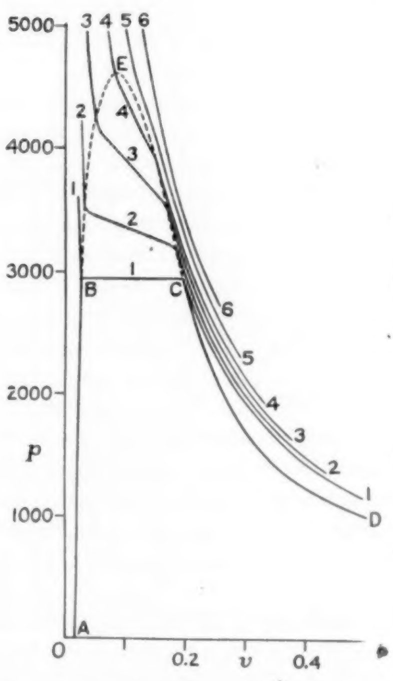


FIG. 3 THE CRITICAL STATE OF WATER

ate temperature, consists of three parts; viewing it as a compression, we have a curve approximately of the form $p\,v = C$ in the superheated range, a horizontal line of condensation across the range of wet steam, and a nearly vertical line for the very slight compression of water by increasing pressure. At each limit line, $A\ B$ and $C\ D$ of Fig. 3, there is an abrupt change of characteristic.

15 Now the critical point as found by Cailletet and Colardeau is the upper limit of a condition which exists throughout the lower

ranges of temperature. It seems reasonable to assume that it is also the upper limit of the other condition just described, or that the complete isothermal numbered 1 in Fig. 3 is the last one that has a horizontal or constant pressure portion. Above *B C* the isothermals must begin to slant upward in crossing the space between the limits marked by extending *D C* and *A B;* only by some such change can they gradually approach the simple continuous curve which must exist for very high temperatures. The changes at the limits *B E* and *C E* probably become less abrupt, or less strictly localized, as the temperature is higher, but there must be some such locus of these changes as is here traced in the outline *B E C*.

16 The condition $c_p = \infty$, which means that heat can be imparted to a substance at constant pressure without raising the temperature, is characteristic of the steam and water mixture whose mechanical condition is represented by any point within the enclosure *A B C D*. Above *B C*, c_p has a finite but gradually decreasing value, since a change of state at constant pressure across the area *B E C* from left to right involves an increase in temperature. To the left of *B E* the substance is liquid, or is in a state of close conglomeration which is analogous to the definite liquid condition farther down; and along *B E* the specific heat will probably continue to increase somewhat after the manner of curve *D* on Fig. 2. The last (highest) isothermal that just touches the region *C E B* will be tangent somewhere below *E*, and it seems reasonable that the highest value of the specific heat just along the outside of *C E B* should be at about this highest temperature.

17 The idea that $c_p = \infty$ at the critical point is based on the assumption that the isothermals across *B E C* are horizontal clear up to the vanishing point; then at the rounded apex *E* we should have an instantaneous condition of constant pressure and constant temperature, which would satisfy the physical meaning of the mathematical expression. The discussion just completed shows that the highest temperature along the curve *B E C* is probably below *E*, toward *C*, and that in any case, the point *E* will be far above the limit of the true liquid state at *B*.

CONCLUSIONS AS TO INITIAL SPECIFIC HEAT

18 Without indulging in further speculation as to the behavior of water above the limit marked by the line *B C* of Fig. 3, we shall now resume consideration of the results of Knoblauch and Jakob, and take up the question whether they really are correctly interpreted,

close to saturation, by the curves drawn by the authors. The method of these experiments, briefly stated, was as follows: The current of steam was first superheated to a desired degree in a preliminary electrical heater, then further raised in temperature by a measured supply of electrical energy in a second heater which really constituted the determining apparatus of the experiment. The result obtained was the mean specific heat over the range from t_1 at entrance to t_2 at exit of the final heating coil. The published tabular results are replotted on Fig. 4, where each circled point shows a value of the mean specific heat, while the short horizontal line marks the range

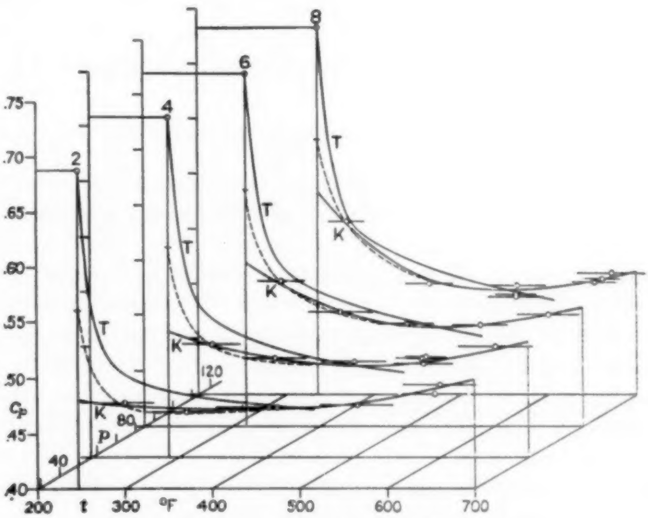


FIG. 4 CLOSE COMPARISON OF CURVES c_p

over which the determination extended. The four original curves, as published, are drawn in full line and marked K; with them are drawn also curves from Thomas' results, carefully interpolated on Fig. 1 for the pressures 2, 4, 6 and 8 kg. and marked T. The lowest determination by Knoblauch and Jakob began at from 12 to 18 deg. fahr. above saturation.

19 Now the dotted curves in Fig. 4, drawn by the writer, show how without putting the least strain upon the actual experimental results, the Knoblauch and Jakob curves of c_p can be made to take a form closely similar to that of Thomas' curves. The saturation line of c_p thus obtained is laid out in the curve marked H_1 on Fig. 2, where

it easily runs into the extended T curve. The writer believes that this is the correct interpretation of the results of Knoblauch and Jakob; and with this interpretation the two sets of experiments strongly confirm each other.

20 Finally, taking what seems to be the most probable mean between the curves T and H_1 over the range of experiment, and further considering that if the curves of c_p and of c_w (the latter the D curve for water) are to meet somewhere beyond the critical ordinate C, the former must be lower at this ordinate because of its more rapid upward slant, the writer has laid out the curve H_2 on Fig. 2 as representing, in his judgment, the truest location that can be made, from present data, of the curve of c_p for zero superheat. This is used as the starting point in the final layout of c_p in Fig. 6 and 7.

THE HIGHER RANGE OF SUPERHEAT

21 As remarked in Par. 7, the two sets of curves show the beginning of a marked divergence toward the higher ranges of superheat. Most of the available data for these ranges have been obtained by the explosion method. This consists in enclosing a combustible mixture of gas (hydrogen) and air (or oxygen) in a vessel, preferably of spherical form, starting ignition at the center of the vessel so that a spherical flame-cap, propagated at a uniform rate, will reach the whole containing surface at the same time, and measuring the resulting instantaneous high pressure before there is a chance for the heat to be dissipated by radiation. This leads to the specific heat at constant volume, from which that at constant pressure can be derived by means of the fundamental physical relation between the two, $c_p - c_v =$ const. The results obtained, all expressed as rectilinear or first-degree functions of the temperature, exhibit rather a wide divergence; but Knoblauch and Jakob show, at the end of their paper, how the assumption that both the mean and the instantaneous values of c_p increase with t according to a curve relation, or at an increasing rate lead to a very close reconciliation of these conflicting indications. Further the curve which they draw agrees very well with the one set of experiments at constant pressure, carried out to quite a fairly high temperature, that is now available.

22 These experiments were made by Holborn and Henning at the *Physical Technischen Reichsanstalt*, and were published in *Annalen der Physik*, vol. 18, 1905, p. 739. Steam at the pressure of the atmosphere was electrically superheated, then passed into a small

absorption calorimeter of the usual type, but with oil instead of water as the heat absorbent. In the four sets of experiments the upper temperature was 270 deg., 440 deg., 620 deg., and 820 deg., cent.; and in each case the steam was cooled to about 110 deg., leaving the calorimeter with this small residuum of superheat. The mean specific heat over the range of cooling was the quantity determined, and the four values are plotted on Fig. 5.

23 The first experiment, from 270 deg. to 110 deg. is represented by the rectangle $C\ A\ B\ D$, the height of $A\ B$ showing what the specific heat would be if it were constant over the whole range, and the

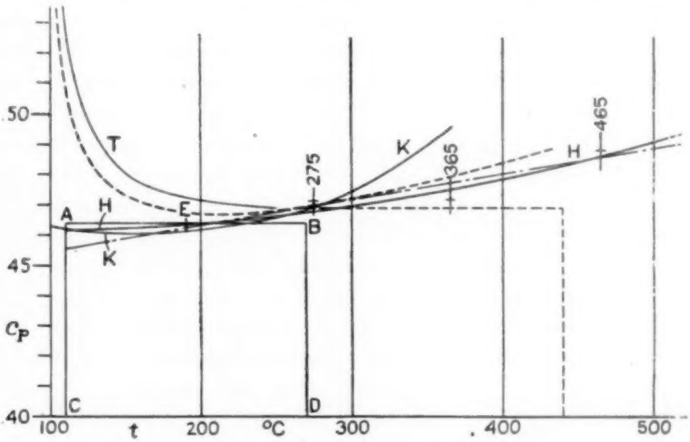


Fig. 5 Specific Heat at Atmospheric Pressure

middle point E locating the mean specific heat at the middle of the range. The three other middle parts, similar to E, are marked by short cross lines on the ordinates at 275 deg., 365 deg., and 465 deg., Now the curve of mean specific heat would go through the outer corners of a series of rectangles like $C\ A\ B\ D$, or through the B points; with rectilinear variation, the line of true specific heat would go through the middle points like E; and a curve of true specific heat should be of such form that its mean height over any given range such as $C\ D$, or the area beneath the curve, will be the same as that of the rectangle like $C A B D$. On Fig. 5 the straight dot and dash line H shows the Holborn-Henning formula; the curve running with this line has been traced by the writer as probably a truer interpretation of the results.

24 The practical application of this last discussion is made evident when we draw in the specific heat curves of Knoblauch and Jakob and of Thomas for the same (atmospheric) pressure, designating them by K and T as heretofore. It appears that the K curve rises much too rapidly, while the T curve fails to show the incipient upward tendency which it ought to have.

FINAL DETERMINING CURVES

25 Using this combination of data as a basis of judgment, and bearing in mind the modification of the K curve made on Fig. 4, the writer has traced in the dotted curve shown on Fig. 5 as representing his conclusion in regard to the most probable value of c_p at atmospheric pressure. Near saturation there is a considerable difference between the T curve and the other two, although the disagreement is exaggerated by the large vertical scale of the drawing. The difference exists in both the form of the curves and their absolute height. The H curve as drawn agrees very well with the original K curve, but there is no reason why a rise toward saturation may not be completely masked in the determination of the heat given off by the steam through a wide range of cooling. This H curve might easily, and with equally close conformity to the experimental data, be replaced by one of our final form, after the same method that was applied to the K curves on Fig. 4. The quantitative discrepancy is less easily overcome; neither set of experiments presents a predominating claim to confidence, and it appears that the best way to arrive at a conclusion is to "split the difference," noting at the same time that, at the most, this difference is not very great either absolutely or relatively.

26 The guiding curves used by the writer in finally laying out c_p on Fig. 7 are drawn on one plane in Fig. 6, for the pressures 15, 100, 300 and 600 lb. per square inch. These derived curves are in full line, marked H, while the corresponding K and T curves are dotted. At 15 lb. there is the discrepancy just discussed. At 100 lb. the agreement is much better, which is all the more satisfactory because this is within the range of ordinary technical application in the steam plant. At 300 lb. very little weight is given to the widely extrapolated K curve. The H curve at 600 will give a much "fairer" surface in Fig. 7 than would the T curve.

RESULTS

27 In Fig. 7 and Table 1 are set forth the final results of the discussion. The diagram, which shows c_p, is laid off in the same general manner as Fig. 1 and 4, but the temperature abscissa is degrees of superheat t_s, instead of temperature on the thermometer scale. Then the cross curves, which give the variation of c_p with change in pressure, are in vertical planes instead of lying on vertical curved surfaces as in Fig. 1. The full line curves cover the whole range, from

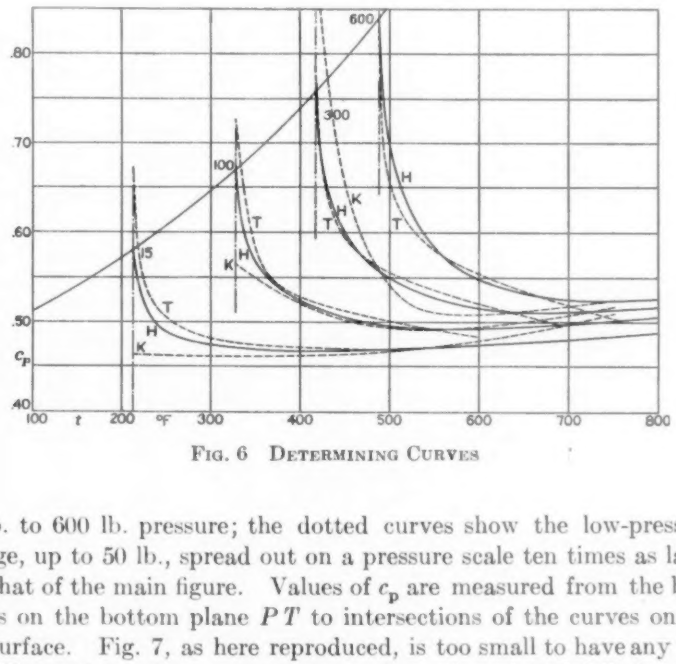


FIG. 6 DETERMINING CURVES

1 lb. to 600 lb. pressure; the dotted curves show the low-pressure range, up to 50 lb., spread out on a pressure scale ten times as large as that of the main figure. Values of c_p are measured from the base lines on the bottom plane PT to intersections of the curves on the c_p surface. Fig. 7, as here reproduced, is too small to have any but an illustrative value: measurements from the original full size diagram are given in Table 1.

28 In the table, the quadruple horizontal lines correspond with the TC curves on Fig. 7, giving quantities which belong to an operation of heating at the constant pressure marked at both outer margins: the columns are analogous to the PC cross curves on Fig. 7, showing variation with the pressure for a particular range of superheat as marked at the top. The four values grouped together for each condition are as follows:

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM

p		DEGREES OF SUPERHEAT							
		0	10	20	40	60	80	100	130
1	t	102.0	112.0	122.0	142.0	162.0	182.0	202.0	232.0
	c	.513	.475	.465	.457	.454	.453	.453	.453
	h	(1113.1)	4.9	9.6	18.8	27.9	36.9	46.0	59.6
	n	(1.9890)	.0086	.0167	.0323	.0472	.0615	.0754	.0955
3	t	141.6	151.6	161.6	181.6	201.6	221.6	241.6	271.6
	c	.535	.490	.474	.463	.459	.457	.456	.456
	h	(1125.1)	5.1	9.9	19.2	28.4	37.6	46.7	60.4
	n	(1.8891)	.0083	.0161	.0309	.0451	.0587	.0719	.0910
6	t	170.1	180.1	190.1	210.1	230.1	250.1	270.1	300.1
	c	.553	.502	.484	.471	.466	.463	.461	.460
	h	(1133.8)	5.2	10.1	19.7	29.0	38.3	47.6	61.4
	n	(1.8276)	.0082	.0158	.0303	.0441	.0573	.0702	.0887
10	t	193.2	203.2	213.2	233.2	253.2	273.2	293.2	323.2
	c	.567	.515	.494	.479	.473	.470	.467	.465
	h	(1140.9)	5.4	10.4	20.1	29.6	39.0	48.4	62.4
	n	(1.7832)	.0081	.0157	.0299	.0434	.0565	.0691	.0873
15	t	213.0	223.0	233.0	253.0	273.0	293.0	313.0	343.0
	c	.581	.526	.504	.487	.480	.475	.472	.469
	h	(1146.9)	5.5	10.6	20.5	30.2	39.7	49.2	63.3
	n	(1.7487)	.0081	.0156	.0296	.0430	.0559	.0683	.0862
20	t	227.9	237.9	247.9	267.9	287.9	307.9	327.9	357.9
	c	.591	.535	.513	.494	.486	.480	.476	.473
	h	(1151.5)	5.6	10.8	20.9	30.7	40.3	49.9	64.1
	n	(1,7244)	.0081	.0155	.0295	.0428	.0555	.0678	.0856
25	t	240.0	250.0	260.0	280.0	300.0	320.0	340.0	370.0
	c	.600	.543	.521	.500	.491	.484	.480	.476
	h	(1155.1)	5.7	11.0	21.2	31.1	40.8	50.5	64.8
	n	(1.7058)	.0080	.0155	.0294	.0426	.0553	.0675	.0851
30	t	250.3	260.3	270.3	290.3	310.3	330.3	350.3	380.3
	c	.608	.550	.527	.506	.496	.489	.484	.479
	h	(1158.3)	5.7	11.1	21.4	31.4	41.3	51.0	65.5
	n	(1.6908)	.0080	.0154	.0293	.0425	.0552	.0673	.0848
40	t	267.1	277.1	287.1	307.1	327.1	347.1	367.1	397.1
	c	.620	.560	.537	.516	.505	.479	.491	.485
	h	(1163.4)	5.9	11.3	21.8	32.0	42.0	51.9	66.6
	n	(1.6677)	.0080	.0154	.0292	.0424	.0549	.0670	.0844
50	t	280.9	290.9	300.9	320.9	340.9	360.9	380.9	410.9
	c	.631	.570	.546	.524	.512	.503	.497	.490
	h	(1167.6)	6.0	11.5	22.2	32.5	42.4	52.6	67.4
	n	(1.6497)	.0080	.0153	.0292	.0422	.0548	.0668	.0841
60	t	292.5	302.5	312.5	332.5	352.5	372.5	392.5	422.5
	c	.640	.577	.553	.531	.517	.508	.502	.494
	h	(1171.2)	6.0	11.7	22.5	32.9	43.2	53.3	68.2
	n	(1.6354)	.0080	.0153	.0291	.0421	.0546	.0666	.0838

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM—Continued

DEGREES OF SUPERHEAT										
160	200	250	300	350	400	450	500	600		p
262.0	302.0	352.0	402.0	452.0	502.0	552.0	602.0	702.0	t	
.453	.454	.455	.457	.460	.463	.466	.469	.478	c	1
73.2	91.3	114.0	136.8	159.8	182.8	206.1	229.4	276.8	h	
.1147	.1392	.1680	.1953	.2211	.2458	.2693	.2919	.3344	n	
301.6	341.6	391.6	441.6	491.6	541.6	591.6	641.6	741.6	t	
.455	.456	.457	.459	.462	.465	.468	.471	.479	c	3
74.1	92.3	115.1	138.0	161.0	184.2	207.5	231.0	278.5	h	
.1093	.1326	.1603	.1864	.2112	.2350	.2577	.2795	.3207	n	
330.1	370.1	420.1	470.1	520.1	570.1	620.1	670.1	770.1	t	
.459	.459	.460	.462	.465	.467	.470	.474	.482	c	6
75.1	93.5	116.5	139.5	162.7	186.0	209.4	233.1	280.8	h	
.1065	.1291	.1260	.1815	.2057	.2289	.2512	.2725	.3130	n	
353.2	393.2	443.2	493.2	543.2	593.2	643.2	693.2	793.2	t	
.464	.463	.463	.465	.468	.471	.474	.477	.485	c	10
76.3	94.9	118.0	141.2	164.5	188.0	211.6	235.4	283.5	h	
.1047	.1270	.1533	.1783	.2022	.2250	.2469	.2680	.3080	n	
373.0	413.0	463.0	513.0	563.0	613.0	663.0	713.0	813.0	t	
.468	.467	.468	.470	.472	.475	.478	.481	.489	c	15
77.4	96.1	119.4	142.9	166.4	190.1	213.9	237.9	286.4	h	
.1034	.1253	.1513	.1761	.1997	.2223	.2440	.2649	.3046	n	
387.9	427.9	477.9	527.9	577.9	627.9	677.9	727.9	827.9	t	
.471	.470	.471	.473	.475	.478	.481	.484	.492	c	20
78.3	97.1	120.6	144.2	167.9	191.7	215.7	239.9	288.7	h	
.1026	.1243	.1500	.1745	.1979	.2204	.2419	.2627	.3022	n	
400.0	440.0	490.0	540.0	590.0	640.0	690.0	740.0	840.0	t	
.473	.472	.474	.476	.478	.481	.484	.487	.495	c	25
79.1	98.0	121.6	145.4	169.2	193.2	217.3	241.6	290.7	h	
.1020	.1235	.1491	.1734	.1967	.2190	.2404	.2611	.3004	n	
410.3	450.3	500.3	550.3	600.3	650.3	700.3	750.3	850.3	t	
.476	.475	.476	.478	.481	.483	.486	.490	.498	c	30
79.8	98.8	122.6	146.5	170.4	194.5	218.8	243.2	292.5	h	
.1016	.1230	.1484	.1726	.1958	.2180	.2393	.2599	.2991	n	
427.1	467.1	517.1	567.1	617.1	667.1	717.1	767.1	867.1	t	
.481	.479	.480	.482	.485	.487	.490	.493	.502	c	40
81.1	100.3	124.2	148.3	172.3	196.7	221.2	245.8	295.5	h	
.1010	.1222	.1474	.1714	.1944	.2164	.2376	.2581	.2971	n	
440.9	480.9	530.9	580.9	630.9	680.9	730.9	780.9	880.9	t	
.486	.483	.483	.485	.488	.491	.494	.497	.505	c	50
82.1	101.4	125.6	149.8	174.1	198.6	223.2	247.9	298.0	h	
.1006	.1216	.1466	.1705	.1933	.2152	.2363	.2567	.2955	n	
452.5	492.5	542.5	592.5	642.5	692.5	752.5	792.5	892.5	t	
.489	.486	.486	.487	.490	.493	.496	.500	.507	c	60
82.9	102.4	126.7	151.0	175.5	200.1	224.8	249.7	300.0	h	
.1002	.1211	.1460	.1697	.1924	.2142	.2352	.2555	.2941	n	

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM—Continued

p		DEGREES OF SUPERHEAT							
		0	10	20	40	60	80	100	130
80	t	311.8	321.8	331.8	351.8	371.8	391.8	411.8	441.8
	c	.655	.589	.564	.540	.525	.515	.508	.500
	h	(1177.0)	6.2	11.9	22.9	33.6	44.0	54.2	69.3
	n	(1.6132)	.0080	.0153	.0290	.0419	.0543	.0662	.0832
100	t	327.6	337.6	347.6	367.6	387.6	407.6	427.6	457.6
	c	.669	.599	.573	.547	.531	.521	.513	.504
	h	(1181.9)	6.3	12.2	23.3	34.1	44.6	55.0	70.2
	n	(1.5963)	.0079	.0152	.0289	.0418	.0541	.0658	.0827
130	t	347.1	357.1	367.1	387.1	407.1	427.1	447.1	477.1
	c	.687	.610	.583	.555	.538	.527	.518	.508
	h	(1187.8)	6.4	12.4	23.8	34.7	45.4	55.8	71.3
	n	(1.5769)	.0079	.0152	.0288	.0416	.0537	.0654	.0821
160	t	363.3	373.3	383.3	403.3	423.3	443.3	463.3	493.3
	c	.703	.621	.593	.562	.544	.532	.523	.513
	h	(1192.7)	6.5	12.6	24.2	35.3	46.0	56.6	72.1
	n	(1.5620)	.0079	.0152	.0287	.0414	.0534	.0650	.0816
200	t	381.6	391.6	401.6	421.6	441.6	461.6	481.6	511.6
	c	.720	.634	.604	.571	.551	.539	.529	.518
	h	(1198.3)	6.7	12.9	24.7	35.9	46.8	57.4	73.1
	n	(1.5462)	.0079	.0152	.0286	.0412	.0532	.0646	.0811
250	t	400.9	410.9	420.9	440.9	460.9	480.9	500.9	530.9
	c	.740	.649	.617	.580	.560	.545	.535	.523
	h	(1204.2)	6.9	13.2	25.1	36.5	47.6	58.4	74.3
	n	(1.5309)	.0079	.0152	.0286	.0411	.0530	.0644	.0806
300	t	417.4	427.4	437.4	457.4	477.4	497.4	517.4	547.4
	c	.757	.661	.628	.588	.566	.551	.541	.529
	h	(1206.9)	7.0	13.5	25.6	37.1	48.3	59.2	75.3
	n	(1.5186)	.0080	.0152	.0286	.0410	.0528	.0641	.0803
350	t	432.0	442.0	452.0	472.0	492.0	512.0	532.0	562.0
	c	.773	.672	.638	.595	.571	.556	.545	.533
	h	(1213.7)	7.1	13.7	26.0	37.6	48.9	59.9	76.1
	n	(1.5085)	.0080	.0152	.0285	.0409	.0526	.0639	.0799
400	t	444.9	454.9	464.9	484.9	504.9	524.9	544.9	574.9
	c	.788	.682	.646	.601	.576	.560	.548	.536
	h	(1217.7)	7.3	13.9	26.3	38.1	49.4	60.5	76.7
	n	(1.5001)	.0080	.0152	.0285	.0408	.0525	.0636	.0795
500	t	467.4	477.4	487.4	507.4	527.4	547.4	567.4	597.4
	c	.815	.701	.658	.609	.582	.565	.552	.539
	h	(1224.5)	7.5	14.3	26.9	38.8	50.3	61.5	77.8
	n	(1.4863)	.0081	.0153	.0285	.0406	.0521	.0631	.0788
600	t	486.9	496.9	506.9	526.9	546.9	566.9	586.9	616.9
	c	.840	.718	.668	.614	.586	.568	.555	.542
	h	(1230.5)	7.7	14.6	27.4	39.4	50.9	62.2	78.6
	n	(1.4757)	.0081	.0153	.0284	.0404	.0518	.0626	.0781

TABLE 1 THERMAL PROPERTIES OF SUPERHEATED STEAM—Continued

DEGREES OF SUPERHEAT										p
160	200	250	300	350	400	450	500	600		
471.8	511.8	561.8	611.8	661.8	711.8	761.8	811.8	911.8	t	
.494	.490	.490	.491	.494	.497	.500	.504	.511	c	80
84.2	103.9	128.4	152.9	177.6	202.4	227.3	252.4	303.1	h	
.0995	.1202	.1447	.1682	.1907	.2123	.2331	.2532	.2916	n	
487.6	527.6	577.6	627.6	677.6	727.6	777.6	827.6	927.6	t	
.498	.493	.493	.495	.497	.500	.503	.507	.515	c	100
85.2	105.1	129.7	154.4	179.2	204.1	229.2	254.4	305.5	h	
.0989	.1193	.1437	.1669	.1892	.2107	.2313	.2513	.2895	n	
507.1	547.1	597.1	647.1	697.1	747.1	797.1	847.1	947.1	t	
.502	.497	.496	.498	.500	.504	.507	.511	.519	c	130
86.4	106.4	131.2	156.1	181.0	206.1	231.4	256.8	308.3	h	
.0980	.1183	.1423	.1653	.1873	.2086	.2291	.2489	.2868	n	
523.2	563.2	613.2	663.2	713.2	763.2	813.2	863.2	963.2	t	
.506	.501	.499	.501	.504	.507	.511	.514	.522	c	160
87.4	107.5	132.5	157.5	182.6	207.9	233.3	258.9	310.7	h	
.0973	.1174	.1412	.1640	.1859	.2070	.2273	.2470	.2848	n	
541.6	581.6	631.6	681.6	731.6	781.6	831.6	881.6	981.6	t	
.511	.505	.504	.505	.508	.511	.514	.518	.526	c	200
88.6	108.9	134.1	159.3	184.6	201.1	235.7	261.5	313.7	h	
.0967	.1166	.1402	.1628	.1845	.2054	.2256	.2452	.2828	n	
560.9	600.9	650.9	700.9	750.9	800.9	850.9	900.9	1000.9	t	
.517	.510	.508	.510	.512	.515	.519	.523	.531	c	250
89.9	110.4	135.9	161.3	186.7	212.5	238.4	264.4	317.1	h	
.0961	.1159	.1393	.1617	.1832	.2040	.2241	.2436	.2809	n	
577.4	617.4	667.4	717.4	767.4	817.4	867.4	917.4	1017.4	t	
.521	.514	.512	.513	.515	.519	.522	.526	.534	c	300
91.0	111.7	137.4	163.0	188.7	214.6	240.6	266.8	319.8	h	
.0956	.1152	.1385	.1607	.1821	.2028	.2228	.2421	.2793	n	
592.0	632.0	682.0	732.0	782.0	832.0	882.0	932.0	1032.0	t	
.542	.517	.515	.516	.518	.522	.525	.529	.537	c	350
91.9	112.8	138.6	164.4	190.2	216.2	242.4	268.7	322.0	h	
.0952	.1146	.1377	.1598	.1811	.2016	.2215	.2407	.2777	n	
604.9	644.9	694.9	744.9	794.9	844.9	894.9	944.9	1044.9	t	
.527	.520	.518	.519	.521	.525	.528	.532	.540	c	400
92.7	113.6	139.6	165.5	191.5	217.6	243.9	270.4	324.0	h	
.0947	.1140	.1370	.1589	.1801	.2005	.2203	.2394	.2763	n	
627.4	667.4	717.4	767.4	817.4	867.4	917.4	967.4	1067.4	t	
.530	.523	.521	.522	.525	.529	.532	.537	.545	c	500
93.9	115.0	141.1	167.2	193.4	219.7	246.3	273.0	327.1	h	
.0938	.1128	.1355	.1572	.1782	.1984	.2180	.2371	.2737	n	
646.9	686.9	736.9	786.9	836.9	886.9	936.9	986.9	1086.9	t	
.534	.527	.525	.526	.528	.532	.536	.540	.549	c	600
94.7	116.0	142.3	168.5	194.9	221.4	248.1	275.0	329.4	h	
.0929	.1117	.1341	.1556	.1764	.1964	.2159	.2348	.2711	n	

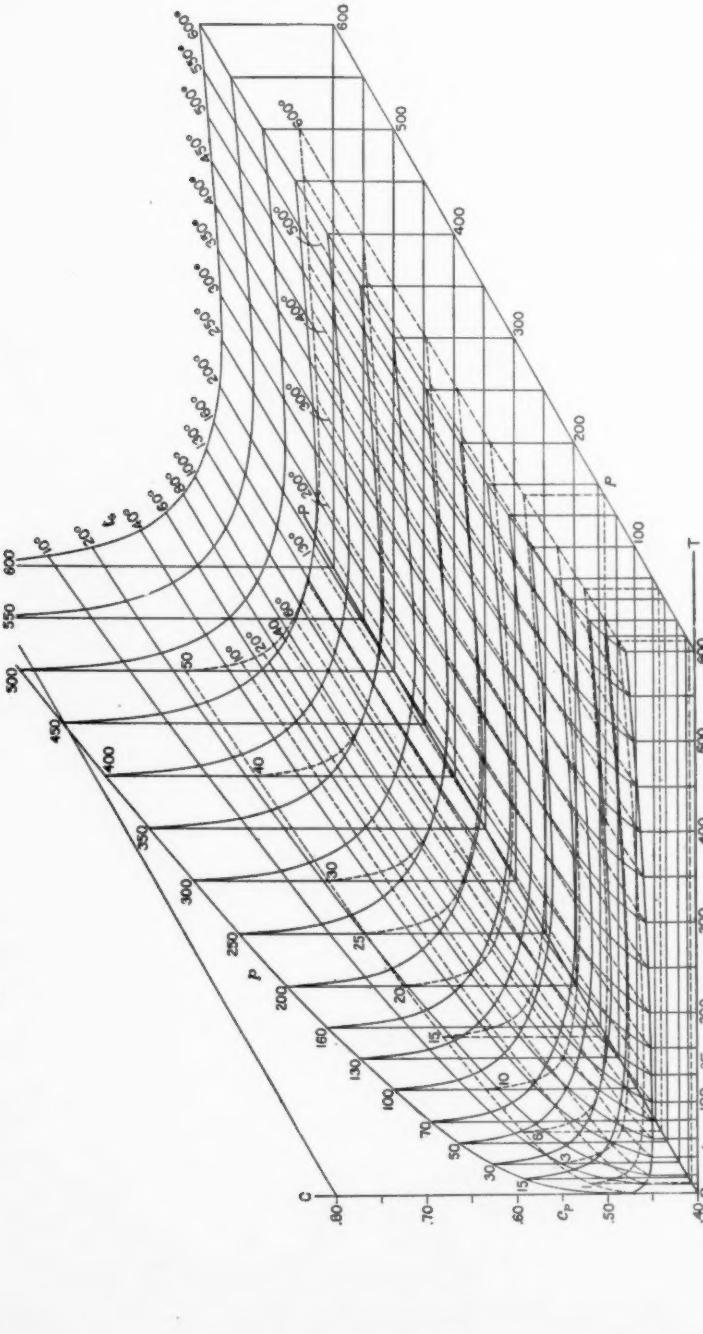


FIG. 7 PROBABLE TRUE SPECIFIC HEAT

t = temperature of steam, in degrees fahrenheit.

c = specific heat under constant pressure, generally called c_p, at the particular temperature and pressure.

h = heat required to raise the steam from saturation up to the the temperature t, or through the number of degrees at the top of the column, at constant pressure.

n = entropy of superheating, or the entropy gained with the heat h.

In choosing the intervals between the tabular values, in both directions, the idea was to keep these intervals short enough for the effective use of simple rectilinear interpolation. A column for 550 deg. was originally computed, also lines for $p = 450$ and 550 lb.; these were crowded out in bringing the table to page form, but their absence will cause no appreciable error.

29 The above description does not fully apply to the first column, for zero superheat. This is, of course, the state of dry saturation, where h and n, as just defined, are both equal to zero. In the spaces that would thus be left vacant are bracketed: In the h column, the total heat H of one pound of dry saturated steam, above water at 32 deg. fahr.; in the n column, the total entropy of heating and evaporation, corresponding with the heat H.

30 The writer, desiring to dispense with Greek-letter symbols in thermodynamic formulae, and having other uses for the more obvious E, has adopted N as the symbol for entropy. Letting plain N stand for the entropy of dry saturated steam formed in the usual manner, which is given in the zero column of the table as just stated, we have $(N + n)$ as the abscissa of a point on the isopiestic curve for superheat in the entropy-temperature system of representation.

31 Fig. 7 embodies the result of an attempt to develop a general law from experimental data which are not in very close agreement. The quantity sought does not lend itself to easy and precise determination; and although the experiments here discussed are essentially correct in method, and are far superior in reliability to any which have preceded them, they nevertheless leave an area of uncertainty within which probabilities must be balanced and judgment exercised. Based upon such data, the law may be qualitatively correct, and yet be liable to a certain degree of quantitative error. But even though the absolute values may not be quite in conformity with the undiscovered truth, it is highly important that in their manner of variation they follow some smoothly acting law.

32 The general principles just enunciated are intended to bear upon the question of the degree of precision desirable in the quantities in Table 1. It appears that there must be some definite law of relation between c, t and p, and there is no reason to expect irregularities in the true law. Whether Fig. 7 embodies this true law or not, there is no doubt that a set of curves defines a quantity much less rigorously than does a mathematical formula. It must be acknowledged therefore that the values of c in Table 1, measured from a diagram and accurate to only three figures, are likely to show small irregularities.

33 The numbers h and n were at first computed directly from c as measured, being carried to five-figure accuracy; that is, the heat values had two decimal places, the entropy values five. Checking these by means of their differences, minor irregularities were found: but by the expedient of making the differences conform to a scheme of regular variation, the values of h and n were brought to a much surer and more exact manner of variation than that of c. Of course, h and n are the actually important quantities for thermodynamic calculations.

34 Although heat and entropy were both worked out to one more decimal place than is given in the table, and together adjusted until the probable irregularity did not exceed two points in this last place, it did not appear that any practically useful purpose would be served by printing the table with more figures than are actually given. In computing the entropy h, values were at first got by plain division, using intervals of 10 to 50 deg., and dividing heat added by average absolute temperature; this first table was then corrected by using more exact methods along a few equally spaced pressure lines, and carrying the differences thus found through the ranges between these closer determinations.

VARIOUS DATA

35 In the preceding discussion, the experiments of Knoblauch and Jakob and of Thomas are accepted as reliable, but no attention is paid to the other experiments, over similar ranges, that have been reported from time to time. Both the "accepted" sets were made by the method of superheating the steam with heat derived from a measured electric current. Knoblauch and Jakob took steam at a temperature t_1 (which ranged from 15 deg. to 370 deg. fahr. above saturation) and raised it to t_2 (which was from 50 to 140 deg., higher than t_1). Thomas found in every case the heat needed to raise the

steam from dry saturation to the particular temperature t. Both took great precautions against radiation, and used effective methods in measuring the unavoidable loss by radiation.

36 The converse of the method of heating is that of cooling, in which heat is abstracted from a current of highly superheated steam and measured by an absorption calorimeter—the steam being still superheated when it leaves the calorimeter. The most prominent experiments along this line were made by Lorenz, and published in the *Zeitschrift des Vereins deutscher Ingenieure*, 1904, p. 698. The surfaces were water-cooled, hence at a much lower temperature than the steam which flowed over them. The results as to c_p now appear to have been uniformly too large, because of a fact which has come to be recognized only in the last few years. This is, that a current of superheated steam may be far from homogeneous, especially when near to saturation. In the method of Lorenz the outer layers of the steam current may be cooled to saturation and even partly condensed, while the body of the steam is yet superheated and determines the reading of the thermometer. The result is that the amount of heat abstracted is too great in relation to the apparent temperature drop.

37 The other method which has been extensively used is that of throttling or wire drawing. The apparatus is essentially a throttling calorimeter on a large scale, with especial precautions against radiation and conduction of heat. The best experiments have been made by Grindley, Philosophical Transactions of the Royal Society of London, 1900, vol. 194, and by Griessmann, *Zeitschrift des Vereins deutscher Ingenieure*, 1903, p. 1850. Two main objections lie against this method, one is the difficulty of insuring that truly dry steam is received at the upper pressure; the other, the uncertainty as to whether the throttled and superheated steam is truly homogeneous. In regard to the latter point, Griessmann, who used a porous plug instead of an orifice, would appear to be less liable to error. The writer analyzed Griessmann's results several years ago; and while the individual values of c_p for the same pressure and nearly the same range of temperature do not agree well, the averages of a number of similar determinations check up quite fairly with values from Table 1, as appears in Table 2. Referring to Fig. 6, we see that Griessmann's results would agree better with the curves of Thomas, which are uniformly higher than those decided upon by the writer, at these low degrees of superheat.

38 In this connection, it may be of interest to compare values of the superheat h as given by Thomas with values from Table 1. The

TABLE 2 COMPARISON OF GRIESSMANN'S RESULTS

Pressure		Degree fahr. of Superheat	Specific heat c_p	
Kg.	Lb.		Griessmann	Table 1
1	14	70	0.51	0.487
2	28	45	0.53	0.503
3	43	30	0.55	0.530
4	57	20	0.57	0.550
5	71	15	0.597	0.570
6	85	10	0.62	0.592

values marked T in Table 3 are taken from Fig. 17 of Thomas' paper, which is a plot of the heat quantity h, those marked H are from Table 1. Neither in absolute amount nor in percentage are the differences large enough to be of much practical significance.

TABLE 3 COMPARISON OF SUPERHEAT IN BRITISH THERMAL UNITS

Pressure	Degree of Superheat							
	40		100		200		300	
P	T	H	T	H	T	H	T	H
20	21.8	20.9	51.3	49.9	99.1	97.1	146.1	144.2
60	23.4	22.5	54.0	55.0	103.6	105.1	151.6	151.0
300	25.2	25.6	59.0	59.2	112.5	111.7	161.7	163.0

SUMMARY

39 For a purpose such as the determination of the total heat of formation of superheated steam in the work-up of a boiler test, the results of either Knoblauch and Jakob or of Thomas might be used, within and over the range of actual experiment, without introducing an error of any practical significance. For other purposes, such as calculations from point to point in the field of superheat, the disagreement of the two authorities introduces a very considerable element of uncertainty. The writer has endeavored to balance probabilities between the two, to correct certain manifest errors in interpretation, and to make the law for c_p take a consistent form, in accordance with the rational requirements which have gradually become evi-

dent as experimental knowledge in regard to this subject has been enlarged.

40 It is reasonable to believe that the change from the properties of wet steam to those of superheated steam, at the saturation limit, is not absolutely abrupt, but that there is left a little disgregation work, or something closely analogous thereto. This accounts for the higher value of c_p near saturation, as well as for its falling off with the temperature. The writer does not believe that the finally adopted initial cross-curve H_2 in Fig. 2 is at all definitely determined, but he does agree on principle with the experiments of Thomas, which show this rise of c_p toward saturation to persist, although decreasing in amount, into the low ranges of pressure and temperature. An error of 10 or even 20 per cent in the value of the initial c_p would have only a minute effect upon the superheat h.

41 The rise of c_p with temperature, after a minimum has been reached, is a well established fact in the behavior of chemically compound gases, like H_2O and CO_2, and even takes place, though at a very slow rate, with the simple gases such as oxygen and nitrogen. Knoblauch and Jakob fully recognize this fact, indeed over emphasize it in letting rather faint experimental indications make their curves rise too soon and too rapidly. Thomas, on the other hand, ignores it, although his experiments hardly extended far enough into the field of superheat for the upward tendency to have come into action. According to the data on Fig. 5, it appears that the writer's curves are made to rise just a little too rapidly. In laying out Fig. 6 and 7, the idea was that these curves of c_p for different pressures, based upon actual temperature and if projected upon one plane as in Fig. 6, ought to converge very slowly as they rise.

42 In conclusion, the writer would again express his strong confidence in the experiments which form the basis of this discussion; and, at the risk of accusation of rashness, would state the belief that, for all technical purposes, the question of the specific heat of superheated steam under constant pressure is about settled.

DISCUSSION

Dr. Henry T. Eddy[1] In considering the comparisons made by Professor Heck of the experimental results arrived at by Professor Thomas, and Knoblauch and Jakob, as referred to by him, the present

[1] Dr. Henry T. Eddy, Dean of the Graduate School and Professor of Mathematics and Mechanics, University of Minnesota, Minneapolis.

writer was led to construct the lines expressing the relation of the total superheat in degrees to the total superheat British thermal units at the four pressures investigated by Knoblauch, viz: at 2, 4, 6 and 8 atmospheres nearly, or 28.44, 56.88, 85.32, and 113.76 lb. per sq. in. precisely, these lines being suitably interpolated to accord with the results of Thomas as given by him in Fig. 9 of his paper published in Transactions, Vol. 29. The lines so drawn would consequently in general lie between those shown by Thomas in his Fig. 17. It then appeared that Professor Thomas had taken some liberties with his original data as shown in his Fig. 9, evidently in order to bring his Fig. 17 into agreement with the idea that saturated steam must begin to be superheated as soon as any heat is imparted to it. That may be the fact and yet the lines on Fig. 17 may be drawn to accord more closely with the data shown in Fig. 9.

2 The object of the present discussion is not to make the slight corrections suggested by these not unwarrantable liberties which Professor Thomas has taken with his data, but to show that his data admit equally well, or in fact better, of a more simple representation which may be regarded as a very close practical approximation to the laws for superheated steam, or steam gas, just as the laws for so-called perfect gases are applied to ordinary gases and are in fact sufficiently close for almost all practical purposes.

3 When we take Thomas' Fig. 9 and attempt to construct his Fig. 17 as he has done, it is found that perfectly straight lines on Fig. 17 represent the mean values of points plotted on Fig. 17 quite as well as the slightly curved lines he has chosen; and had it not been for the supposed necessity for making the lines pass through the origin, no one would have drawn them as they are shown in Fig. 17. Furthermore, it is found by trial that using fahrenheit degrees and British thermal units as coördinates these lines all pass very nearly through a single point whose coördinates are −34 deg. fahr. and −14.5 B.t.u. from saturation as origin; so that Thomas' experiments may all be very closely represented by the linear equation

$$h = C_p(d + 34) - 14.5$$

in which

h = the superheat in British thermal units.
d = the superheat in degrees fahrenheit.
C_p = the specific heat at the given pressure p.
p = the absolute pressure in pounds per square inch.

4 In order to compare this equation with the experiments of

Thomas, let us first compute by it the values of C_p for the data given in Thomas' Fig. 9, and also given in his Table 1 where h and d have been changed from watts and centigrade degrees to British thermal units and degrees fahrenheit by making 1 watt = 3.412 B.t.u., etc.

5 Considering now the values of C_p for these six different pressures, there are certain reasons which will appear later for assuming the mean value of C_p for any pressure to be the average of the middle four temperatures. We shall then provisionally assume them to be independent of the degree of superheat as follows: Let these values be plotted as shown in the small circles in Fig. 1. They fall very closely upon a regular curve, except at 165 lb. From this curve then, they may be read as a systematic set of corrected values of C_p for the above pressures or any other intermediate pressures desired.

TABLE 1 CALCULATED RESULTS FOR COMPARISON WITH THE VALUES OF THOMAS

p	d	h	C_p	p	d	h	C_p
	36	19.65	0.4880		36	21.5	0.5143
	72	36.2	0.4780		72	41.3	0.5264
7	108	54.5	0.4862	115	108	59.36	0.5202
	144	69.6	0.4724		144	79.6	0.5286
	180	87.6	0.4772		180	97.7	0.5243
	270	130.3	0.4763		270	140.9	0.5112
	36	20.33	0.4980		36	22.23	0.5248
	72	37.5	0.4906		72	42.3	0.5359
20	108	55.3	0.4914	165	108	62.1	0.5394
	144	71.3	0.4820		144	82.23	0.5434
	180	88.7	0.4822		180	101	0.5397
	270	132.9	0.4832		270	143	0.5181
	36	20.5	0.5000		36	22.5	0.5286
	72	38.55	0.5000		72	41.6	0.5292
35	108	56.64	0.5100	215	108	62.1	0.5394
	144	72.9	0.4910		144	81.9	0.5416
	180	91.1	0.4935		180	101.55	0.5421
	270	134.1	0.4888		270	145.1	9.5250
	36	20.8	0.5043		36	22.9	0.5343
	72	39.6	0.5104		72	43	0.5425
55	108	58	0.5105	300	108	63.2	0.5472
	144	75.4	0.5051		144	82.9	0.5472
	180	92.8	0.5014		180	102.7	0.5467
	270	136.5	0.4967		270	149.7	0.5300
	36	21.15	0.5009		36	22.9	0.5343
	72	40.25	0.5165		72	45	0.5613
75	108	58.1	0.5113	500	108	64.5	0.5563
	144	76.4	0.5163		144	83.9	0.5528
	180	95.2	0.5126		180	104.75	0.5572
	270	138.6	0.5026		270	150.8	0.5438

TABLE 2 VALUES OF C_p CORRESPONDING TO PRESSURES IN TABLE 1

p	7	20	35	55	75	115	165	215	300	500
C_p	0.478	0.486	0.496	0.507	0.514	0.525	0.54	0.538	0.546	0.557

6 Upon Fig. 1 are plotted, not only the values of p and C_p in Table 2, but also the values of d and C_p given in Table 1, the points for any given pressure being joined by dotted lines to show the deviations of the experimental values of C_p from the assumed mean values, which last are represented by horizontal unbroken lines. These show: First, that the experimental deviations from the mean values adopted follow no general fixed law whatever, such deviations here being represented on an enormously exaggerated scale by reason of the

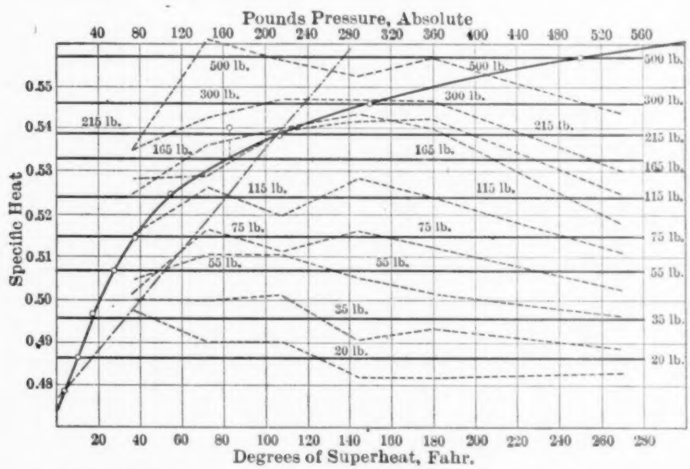


FIG. 1 THE CURVE OF PRESSURE AND SPECIFIC HEAT. THE DEVIATIONS OF THOMAS' EXPERIMENTAL SPECIFIC HEATS FROM THE MEAN VALUES

zero of specific heat having been placed so far below the diagram. Second, that the values of C_p for 36 deg. fahr. all lie much nearer together than do the mean lines assumed; being, in fact, about one-half as far apart. Were this drawing together also further indicated by Thomas' experiments, at 18 deg. of superheat, one might be led to think that the superheat lines converge more rapidly as they approach zero of superheat, but the experiments at 18 deg. do not bear out this interpretation, and so far as they go they confirm the assumption that the lines are straight.

As, however, the experiments at 18 deg. were made at a set of pressures differing from the other lines, it is impossible to introduce the data for 18 deg. along with those for other superheats. This evidence will appear later in Fig. 2. Third, that the values of C_p at 270 deg. fahr. are all much lower than the means assumed. Fourth, that were it a fact that the superheat lines have the kind of curvature shown in Thomas' Fig. 17, the dotted lines in this Fig. 1 should all show unmistakably a form convex upward, such as might be indicated by the dotted lines at pressures of 165, 215 and 300 lb. Fifth, since corresponding curves derived from Knoblauch's experiments are all very strongly concave upward, and the values indicated by his experiments are for all pressures like the dotted line for 20 lb. in Fig. 1, in which the specific heat decreases at higher superheats, we ought to neglect the doubtful indications of convexity that might be suggested at pressures 165, 215 and 300, especially as the specific heats at 165 are in grave doubt by reason of the anomalous position of some of them so near to or even above those on the 215 line. Sixth, that aside from these there are no general relations discoverable, and that a constant value of C_p, i.e., a horizontal straight line for each pressure, fits these values of C_p as well as any other kind of curve that could be suggested for them all.

7 Since Professor Thomas has, in his Fig. 17, constructed superheat lines for a certain set of pressures, let us proceed to construct lines for these pressures in accordance with our equation of superheat and Fig. 1. In Fig. 2 are drawn superheat lines having as their slopes values of C_p derived from Fig. 1. All these lines pass through A, which may be called the absolute origin of the specific heats of superheated steam regarded as a gas, this origin being situated at -34 deg. fahr. and -14 B.t.u. We read from Fig. 1 the following values of C_p:

TABLE 3 VALUES OF Cp PLOTTED FROM FIG. 1

p	0	20	40	60	80	100	150	300	500
C_p	0.474	0.4866	0.4993	0.5087	0.5165	0.522	0.5308	0.546	0.557

8 Having constructed the lines of superheat on Fig. 2, it is possible to construct a superheat curve for any number of degrees of superheat, the coördinates being absolute pressures and British thermal units of superheat. A set of such curves is drawn upon Fig. 2 for the degrees of superheat which Thomas investigated and they show the British thermal units which must be imparted to each pound

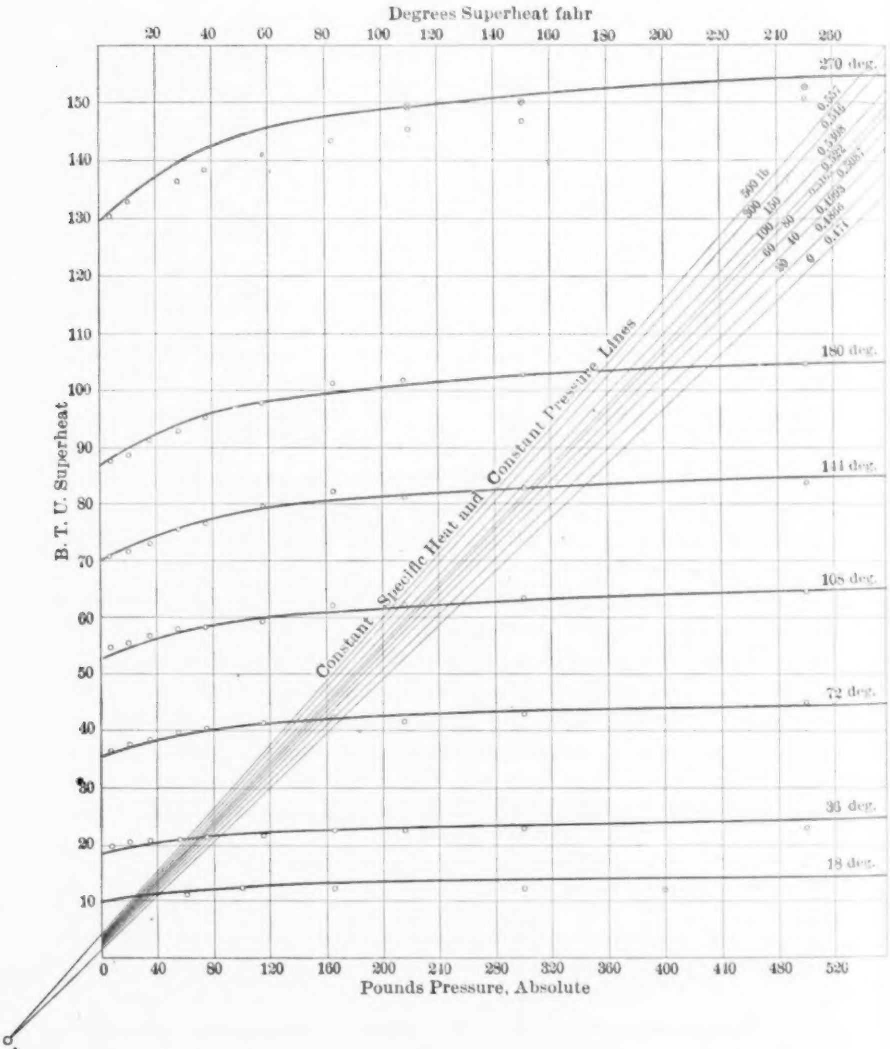


FIG. 2 CURVES OF SUPERHEAT AS PRESSURE AND BRITISH THERMAL UNITS VARY AND OF PRESSURE AND SPECIFIC HEAT AS DEGREES AND BRITISH THERMAL UNITS VARY

of steam to superheat it to the degree indicated, on the supposition that our equation and Fig. 1 correctly represent the facts.

9 These two sets of superheat lines and curves may be best conceived of as profiles of a surface in space, wherein pressures and degrees of superheat are used as coördinates in a horizontal plane and the British thermal units taken for the vertical ordinate. The surface has approximately a plane slope with one edge at A, but is warped enough so that vertical planes parallel to the edge A cut the surface in curves not quite horizontal.

10 We ave also plotted upon Fig. 2 small circles representing tne

TABLE COMPARISON OF RESULTS OF KNOBLAUCH AND THOMAS

		KNOBLAUCH		THOMAS		B.T.U.
p	d	C_p	h	C_p	h	$\triangle h$
28.44	100	0.495	49.5	0.492	51.4	1.9
	200	0.495	99.		100.6	1.6
	300	0.495	148.5		149.8	1.3
	400	0.495	198.		199.	1.
56.88	100	0.50	50.	0.507	53.4	3.4
	200	0.494	98.8		104.1	5.3
	300	0.492	147.6		154.8	7.2
	400	0.492	196.8		205.5	8.7
85.32	100	0.524	52.4	0.518	54.9	2.5
	200	0.51	102.		106.7	4.7
	300	0.505	151.5		158.5	7.
	400	0.504	201.5		210.3	8.8
113.76	100	0.544	54.4	0.525	55.8	1.4
	200	0.522	104.4		108.3	3.9
	300	0.514	154.2		160.8	6.6
	400	0.513	205.2		213.3	8.1

experimental results of Thomas' experiments in his Table 1 and his Fig. 9. The agreement with the theoretical curves is as good as are Professor Thomas' curves in his Fig. 9 for the four middle curves. The divergence is greatest at the curve for 270 deg. In Thomas' Fig. 15 will be found the results of his experiments with his electric calorimeter compared with the double radiation calorimeter which greatly diverge at the 270 deg. curve, though practically coincident elsewhere. I have ventured to place upon my Fig. 2 three of the double radiation results of Thomas, as shown by the crossed circles, which lie much nearer my theoretical curve.

11 It is evident that much greater possibilities of experimental deviations exist at high superheats, and owing to these contradictions and uncertainties I have not included the specific heats derived from the 270 deg. experiments in finding the mean specific heats. This is the same matter that was alluded to in connection with Fig. 1. Regarding the points on the curves at 18 deg. and 36 deg., since the slopes C_p will be much more seriously affected by small deviations in experimental values of points so near A as they are, I have also not included these in computing mean specific heats in Table 2.

12 Now let us consider the fact that the superheat lines of Fig. 2 fail to pass through the origin of superheat. First, if these lines converged so as to meet at $d = 0$, h = constant, we should at once assume the steam to be moist or to contain entrained water. That indeed may be the case even now and it is advisable to make it more certain experimentally that no water is possibly carried along in the superheated steam; second, it is possible that these lines come down to $d = 0$, $h = 0$, very suddenly at saturation, either all at the same slope or at different slopes. No one at present knows what that slope may be, or whether it is a slope or a sudden drop as shown in Fig. 2. Consequently, all discussion as to the value of C_p at saturation, is futile and illusory. Thomas' experiments show no more about that than appears upon this Fig. 2, and Knoblauch did not attempt to obtain the specific heat near saturation. The prolongation of any of these curves to saturation is the merest guess. In fact, finding the value of C_p at saturation is of no practical importance except for the purpose of finding the British thermal units of superheat h, which is represented with approximate accuracy for technical purposes by our equation or by the superheat lines in Fig. 1.

13 It has been thought by the writer that for professional purposes a chart which would map out the superheat region in a different manner from the foregoing and which might replace tables of the properties of superheated steam might be most useful. A part of such a chart is given in Fig. 3 based on the foregoing interpretation of Thomas' experiments, and a more complete chart of the properties of superheated steam based on the experiments of Knoblauch is also appended, as Fig. 4, which was constructed in 1907 by Mr. C. F. Shoop under the writer's directions. Mr. Shoop also constructed a plaster model in the spring of 1907 similar to that represented by the Knoblauch curves in Professor Heck's Fig. 4, which may be seen at the University of Minnesota. The curves of constant specific heat in the chart, Fig. 4, are the level contour lines of the plaster model.

It is evident that using pressures and temperatures as the coördinates to define the state of the superheated steam, it is possible to represent the numerical value of any property such as specific heat, entropy, etc., as ordinates and so get one surface for specific heats, another for entropy, etc. The level contour lines of such surfaces are the lines of constant specific heat, constant entropy, etc., upon the chart, from which can be read numerical values of these quantities for any given pressure and temperature. In fact the numerical values of

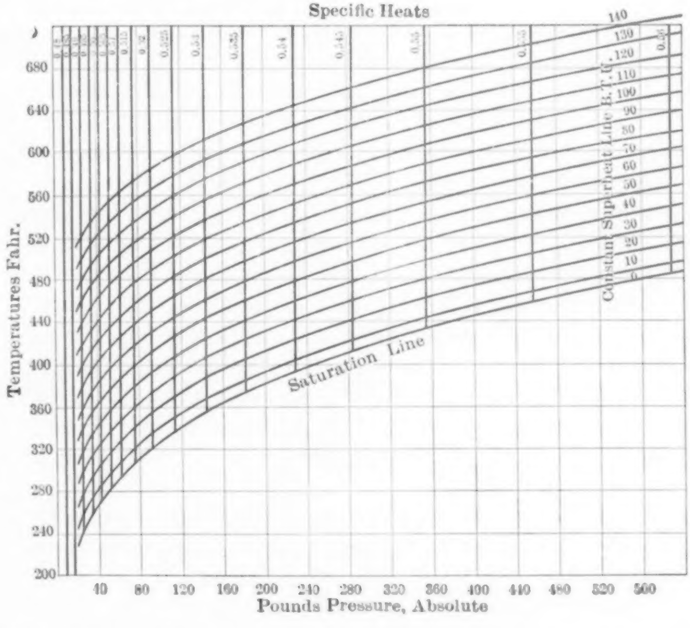


FIG. 3. SPECIFIC HEAT AND BRITISH THERMAL UNITS ABOVE SATURATION OF SUPERHEATED STEAM

any two properties might thus be used as coördinates and the others to constitute surfaces by using them as ordinates. What I wish to advocate is the advantage in point of simplicity of apprehension of using as primary coördinates, pressure and temperature.

14 The curves of constant specific heat and constant mean specific heat up to any given state may be read upon this chart, and the latter may be depended upon as being accurate according to Knoblauch's experiments, having been integrated and checked with great care by Mr. Shoop, to whom I am greatly indebted for the pains-

taking accuracy with which he has executed the computations for this chart. Thus the British thermal units at any given state may be found with confidence from this chart so far as Knoblauch's experiments and interpretations justify such confidence. It should be noticed, however, that Knoblauch's actual experiments cover only a part of the area charted, viz: that from pressure 28.44 to 113.76 and from saturation up to about 675 deg. fahr.

15 On Fig. 3 the lines of specific heat C_p being constant for each pressure are vertical lines, and there are no lines of mean specific heat, since the total superheat in British thermal units is h and is to be computed from our equation when the degree of superheat d is known.

16 It seems desirable to show numerically how Knoblauch's results, smoothed out by equalizing curves as they are at the four pressures he experimented upon, compare with Thomas' results, smoothed out as I have done in Fig. 1 by help of lines whose specific heats are constant at constant pressure, by computing values of h in both cases, as in Table 4, where C_p designates the mean specific heat at constant pressure according to Knoblauch, as found upon the chart in Fig. 4. From Table 4 it is evident that within the range of Knoblauch's experiments his results give smaller amounts of superheat in British thermal units for the same degree of superheat than do Thomas' experiments, and the results differ more the greater the degree of superheat. The difference, however, would not be so great if we accept Thomas' results at 270 deg. of superheat as given in his Fig. 9.

17 Fig. 4 covers the entire range of Knoblauch's experiments. In the part of our chart at higher pressures where values have been exterpolated by Knoblauch from his experiments, his results are also in general smaller than Thomas' by like amounts, except for a short space near saturation, where some of Knoblauch's very high specific heats (over 0.6) make his results larger than Thomas' for small superheats under 100.

SUMMARY

18 Professor Thomas' experiments furnish no sufficient basis for the conclusion that his superheat lines are anything other than straight lines.

19 Those straight lines are represented within the limits of experimental error by the equation,

$$h = C_p (d + 34) - 14.5$$

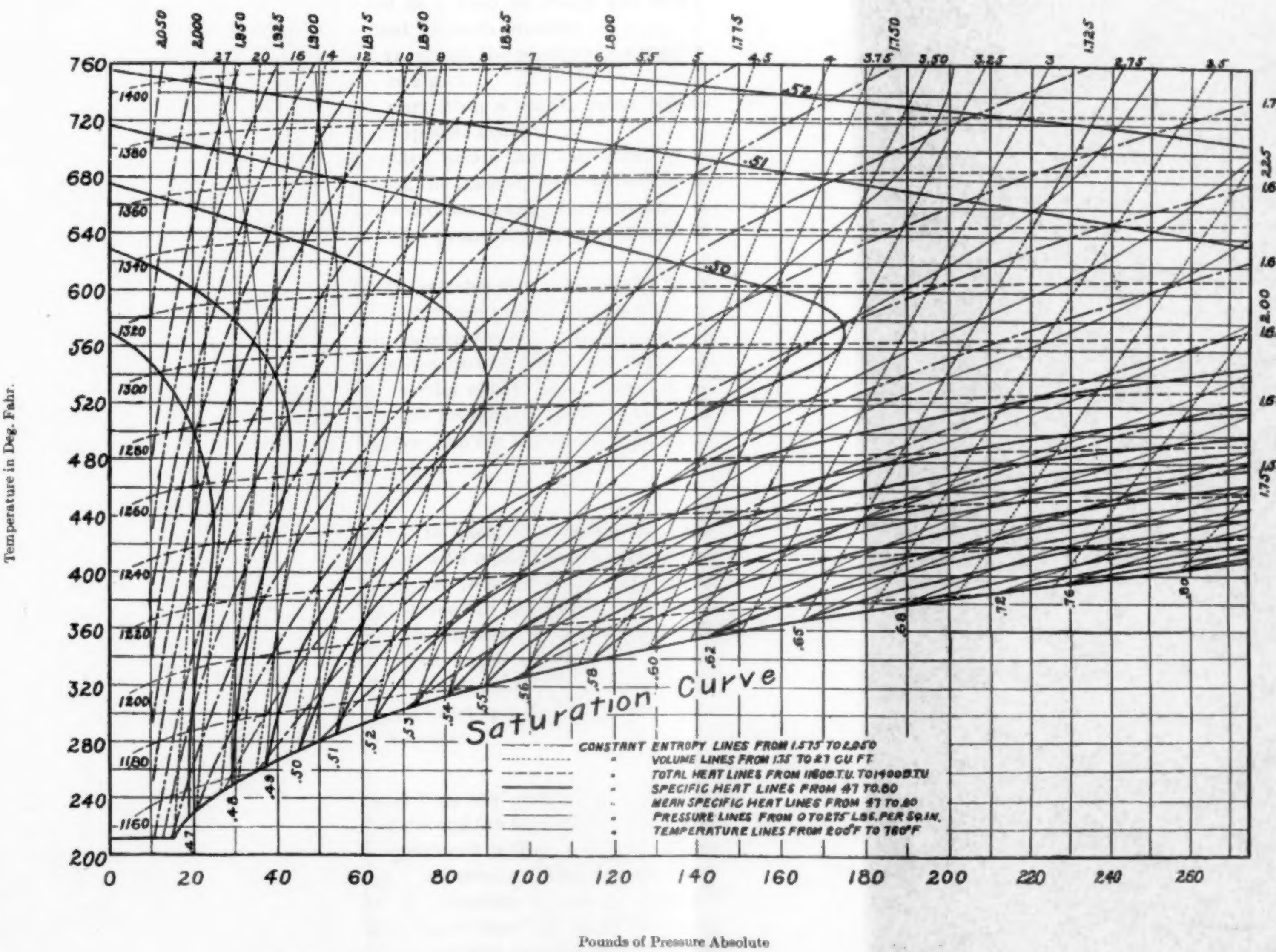


FIG. 4 PROPERTIES OF SUPERHEATED STEAM BASED ON EXPERIMENTS OF KNOBLAUCH AND JAKOB

20 His experiments, therefore, show that superheated steam may for technical purposes be regarded as a kind of steam gas which has a different constant specific heat for each pressure.

21 His constant specific heats vary with the pressure in a regular manner represented by the curve in Fig. 1.

22 The specific heat lines all start from a single point which may be called the absolute origin of specific heats.

23 The properties of superheated steam may be most conveniently tabulated in a chart in which the state of the steam is defined by its pressure and temperature as coördinates, and numbered curves for constant values of each of the other properties which it is desired to tabulate.

24 It would seem to be extremely desirable that Professor Thomas or some other experimenter should introduce a throttling calorimeter into the highly perfected Thomas calorimeter and make investigations that will reconcile the results reached by those two different methods. Could that be done the results would have a conclusiveness now unattainable. By doing this Professor Thomas would add greatly to the debt the profession owes him for the exceedingly valuable researches he has already completed.

Mr. H. H. Suplee Referring to the question of steam gas, this matter is of importance in connection with recent investigations into the operation of gas turbines, or rather those turbines using gases which have had their temperature reduced by the injection of water or steam. In such mixed turbines the temperatures are so high that the steam is superheated to an extent which causes it to become practically steam-gas, and in the computations relating to such work it is especially desirable to give the correct specific heat of the mixture at the high temperatures of operation.

Prof. William D. Ennis Aside from any question as to the relative weights to be assigned the Thomas and the Knoblauch and Jakob experiments, there is a marked agreement within a certain range, (as Professor Thomas has already indicated), within which it is most assuredly safe to say that the question is "about settled." The Thomas experiments covered pressures from 7 lb. to 500 lb., abs., with superheat ranging from zero to a variable upper limit (noted on Fig. 6 and 7 of the description. This limit was in one series of tests, above 780 deg. fahr., the German experiments were made at pressures

of 14.25, 28.5, 57., 85.5 and 114. lb. abs., and at temperatures ranging from a few degrees above saturation up to 400 deg. cent.

2 Considering only the ranges included by both observers, we have the accompanying table, in which no derived values are given, but only those actually obtained by experiment. The italic figures are those of Thomas, obtained by interpolation from his Fig. 6; those in ordinary type are given by Professor Greene, Transactions, Vol. 29, from the Knoblauch and Jakob report. From temperatures 392 to 608, inclusive, the difference in the specific heats as independently determined does not exceed 0.01 B.t.u. excepting in the five cases double-bracketed; and in these five cases the maximum disagreement does not exceed 0.013 B.t.u.

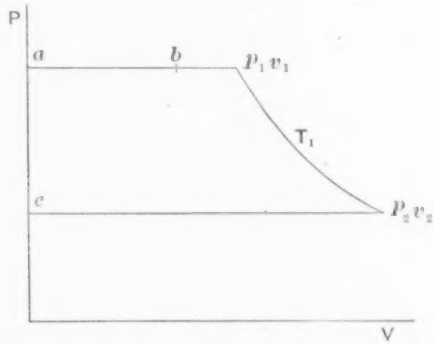


FIG. 1 FOR DEMONSTRATION OF RANKINE'S THEOREM

3 The heavy black lines indicating the range of what may be called the "close agreement zone" broaden out toward the right, the region of higher pressures, in which Professor Thomas' are the only explorations. The high temperature determinations along the 114 lb. pressure column show a constantly increasing amount of divergence between the two sets of experiments. Equally remarkable is the steady increase in amount of divergence, along each pressure column, as the temperature is decreased. It seems reasonable to assume that the limits of accuracy in experiments of this kind would lead to just such results. At low temperatures, the quantity of heat measured is small, and more opportunity exists for errors in calibration and correction, while the value of the constant is known to be excessively variable as we approach the condition of saturation; at high temperatures the radiation losses are greater. In some intermediate region

of temperature, the aggregate of the causes of error falls to a minimum.

4 Why, however, should the coincidence between the two determinations broaden out with increasing pressure? The answer is obvious, that if determinations at higher temperatures and under pressures below 114 lb. had been made by Professor Thomas, there is no reason to believe that the heavy black line marking the limits of close convergence would not have been horizontal along the bottom, as it is along the top. In fact, exterpolation of points on the uniform curves by Professor Thomas gave a very nearly horizontal line, dropping only 18 deg. in temperature at 85.5 lb. pressure, and 18 deg. more at 114 lb.

5 Outside the limits of this agreement, we have two courses before us: either the assignment of prior merit to one or other of the determinations, or an interpretative harmonizing such as Professor Heck presents. On a matter which permits of direct experimental determination, it would seem best to base any final tabulation on the experiments. Professor Heck shows clearly the regions in which confirmatory tests are desirable. The searching analysis to which he submits the few results obtained near those regions may corroborate, but it cannot replace, observation.

6 There is possibly a chance of confusion in the use of the "true" and "mean" values of the specific heat. In the expression for entropy,

$$\phi = \int_{T_1}^{T_2} \frac{\partial H}{T} = C_p \log_2 \frac{T_2}{T_1}$$

the value of C_p must be the *mean* for a temperature range of T_1 to T_2.

7 In Professor Thomas' Fig. 20 (Transactions, Vol. 29) the total heat in superheated steam, calculated from his experiments, is shown not to be independent of the pressure, but to depend on the pressure less directly than does that of saturated steam. At high values, the total heat, while the temperature remains constant, decreases with decrease of pressure for a time, and then increases with decrease of pressure. At low values, the total heat continually increases as the pressure decreases, the temperature remaining constant. Above 1320 B.t.u., there is a definite limited range of pressures throughout which the total heat remains constant at a constant temperature, regardless of the pressure. While the breadth of this pressure zone does not increase as the temperature is increased, the

TABLE 1 VALUES OF THE SPECIFIC HEAT OF SUPERHEATED STEAM

Pressure, lb. per sq. in.	14.25	28.5	57	85.5	114 -
Temperature of saturated steam	210	248	289	316	336
Temperature, deg. fahr.	TRUE SPECIFIC HEATS AT TEMPERATURES STATED				
212	0.463				
230	0.463 *0.502*				
248	0.462 *0.490*	0.480			
266	0.462 *0.483*	0.479 *0.600*			
284	0.462 *0.479*	0.477 *0.523*			
302	0.462 *0.477*	0.476 *0.505*	0.510		
320	0.461 *0.475*	0.475 *0.495*	0.506 *0.568*	0.545	
338	0.461 *0.474*	0.474 *0.490*	0.502 *0.530*	0.536	0.582
356	0.462 *0.473*	0.474 *0.486*	0.498 *0.515*	0.528 *0.553*	0.566 *0.623*
374	0.462 *0.473*	0.473 *0.484*	0.495 *0.508*	0.520 *0.534*	0.552 *0.567*
392	0.462 *0.472*	0.472 *0.482*	0.492 *0.502*	0.513 *0.523*	0.538 *0.545*
410	0.462 *0.472*	0.472 *0.480*	0.489 *0.498*	0.507 *0.517*	0.526 *0.534*
428	0.463 *0.471*	0.472 *0.479*	0.487 *0.495*	0.502 *0.512*	0.517 *0.526*
446	0.464 *0.470*	0.472 *0.477*	0.486 *0.492*	0.497 *0.508*	0.509 *0.521*
464	0.465 *0.469*	0.472 *0.475*	0.485 *0.490*	0.494 *0.503*	0.503 *0.516*
482	0.466 *0.469*	0.473 *0.474*	0.484 *0.487*	0.491 *0.499*	0.499 *0.511*
500			0.483 *0.485*	0.490 *0.497*	0.496 *0.507*
518			0.483 *0.482*	0.489 *0.493*	0.494 *0.503*
536			0.483 *0.481*	0.488 *0.490*	0.493 *0.499*
554			0.484 *0.478*	0.489 *0.487*	0.493 *0.495*
572				0.490 *0.484*	0.493 *0.491*
590				0.491 *0.481*	0.495 *0.489*
608					0.496 *0.485*

TABLE 1 VALUES OF THE SPECIFIC HEAT OF SUPERHEATED STEAM—Continued

Pressure, lb. per sq. in.	14.25	28.5	57	85.5	114
Temperature of saturated steam	210	248	289	316	336
Temperature, deg. fahr.	TRUE SPECIFIC HEATS AT TEMPERATURES STATED				
626	....	....	....	....	0.499 0.483
644	....	....	....	....	0.501 0.478
662	....	....	....	....	0.503 0.477
680	....	....	....	....	0.506 0.474
698	....	....	....	....	0.509 0.473
716	....	....	....	....	0.512 0.471
734	....	....	....	....	0.515 0.499
752	....	....	....	....	0.519 0.468

average departure of the total heat from the mean value, at any temperature, while the temperature varies, appears to grow continually less. To this extent, Rankine's theorem that the total heat is independent of the pressure would seem to be confirmed. Rankine's argument rests upon several assumptions, however, mainly with regard to the constants used. The development is as follows: In Fig. 1, *ab* represents the evaporation of water from *a* to dry steam at b; $b - p_1v_1$ represents superheating to temperature T_1; $p_1v_1 - p_2v_2$ isothermal expansion, $p_2v_2 - c$, cooling and condensation at p_2; *ca* the heating of the water from *c* to *a*. If H_1 = the total heat absorbed along $cab\text{-}p_1v_1$; and H_2 the total heat rejected along $p_2v_2 - c$; we have

$$H_1 + cT_1 \log_e \frac{v_2}{v_1} - H_2 = p_1v_1 - p_2v_2 + cT_1 \log_e \frac{v_2}{v_1}$$

and since

$$p_1v_1 = p_2v_2,$$
$$H_1 = H_2$$

That is, the total heat at p_1v_1 is the same as that at p_2v_2; or in other words depends on the temperature only, and not on the pressure.

The assumptions are, that superheated steam is a perfect gas, following the law $p_1v_1 = p_2v_2 = cT$, and that the mechanical work done during the heating of the water along *ca* may be neglected. The latter assumption may be waived; but the law of expansion of superheated steam is at least doubtful. It certainly diverges far from that of a perfect gas when near saturation.

8 The analysis indicates the necessity for an accurate table of the volumes of superheated steam at various pressures and temperatures. Most practical problems involving superheat also involve the use of figures for specific volume. This problem and that of the specific heat, belong together. When both questions are finally settled, I hope Professor Heck will give us a table of entropies, specific heats, total heats, and volumes that can be used along with the saturated steam table and will be at least as accurate.

Dr. Harvey N. Davis[1] Professor Heck is undoubtedly right in selecting, from a confused literature, the papers of Knoblauch and Jakob and of Thomas as most worthy of consideration, and fortunately there is substantial agreement between them in the region of moderate superheat, even though their curves look so different that one is liable to be deceived on this point.

2 Nevertheless, as Professor Heck has pointed out, there are three respects in which Knoblauch and Thomas are in serious disagreement. First, Thomas' values at low pressures run very much higher close to saturation than do Knoblauch's. Second, Thomas' values at high pressures run lower close to saturation than Knoblauch's. And third, Thomas' curves do not pass a minimum in the region of high superheat and rise again as Knoblauch and others believe they should. Professor Heck's discussion of the last of these points of disagreement is particularly satisfactory, including his conclusion that Knoblauch's curves rise rather too rapidly. The second will be referred to later in a paper on the total heat of saturated steam to be presented to this Society at its December meeting. It will there be shown that what little evidence there is seems to be with Knoblauch against Thomas. The discussion in this note will be confined to the first of the three matters, the contention being that here also Professor Heck should have decided with Knoblauch against Thomas.

3 The point at issue is whether or not the fundamental curve at 15

[1] Dr. Harvey N. Davis, Instructor in Physics and Mathematics, Harvard University.

lb., and those near it, should swoop suddenly to a considerable height close to the saturation line as do Thomas'. Professor Heck believes that they should, and forces through Knoblauch's experimental points curves which are as nearly like Thomas' in shape as possible.

4 Similarly sudden changes in various properties of superheated steam close to the saturation line have been reported several times before. For instance, Grindley found something exactly similar in his wire-drawing experiments, and so did Battelli and also Ramsay and Young in their volume measurements; and in every case of this sort, the trouble has invariably been traced to unevaporated water floating in very minute drops in apparently superheated steam. The form which the explanation takes in this particular case is simple. Thomas starts with admittedly wet steam, and, in a preliminary experiment, undertakes to dry it without any superheating. He believes that he can tell exactly when this has been accomplished by watching for the first jump in his thermo-couple temperature, and justifies this belief by certain control measurements, not described, of the quality of his steam, which measurements would presumably be carried out in a second calorimeter at some distance from the thermo-couple. Every bit of experience hitherto reported by careful observers would lead one to expect that that part of his mixture which is already evaporated would begin to superheat, and the thermo-couple to jump, before the last traces of entrained water had disappeared; and further that evaporation would proceed at the expense of the heat in the steam while the mixture was passing to the place where it was to be examined, so that this examination might wholly fail to detect the wetness of the steam as it was passing the thermo-couple.

5 Knoblauch and Jakob made a careful study of this very point in their electrical preheater, and found experimental proof that water could be held in suspension in very highly superheated steam, and that for the process of evaporation time and thorough mixing were essential, as well as heat. Their apparatus provided for this purpose a complicated grid of wires completely filling a pipe nearly 8 in. in diameter and about 8½ ft. long, with elaborate arrangements for keeping track of the distribution of temperature through both the length and the breadth of the pipe. They state that traces of entrained water were observable through several sections of this pipe, and even if "several" means as few as two, and even if it be assumed that the steam in these sections had always a specific volume as great as that of the still more highly superheated steam in the calorimeter, it is easy to compute that the wetness in their steam persisted

for a time which was never less than a second and averaged more than two seconds, and this, too, after *all* of the heat necessary for the high superheat had been put in.

6 In Thomas' apparatus, on the other hand, all the evaporation there was had to take place in 24 quarter-inch holes in a soapstone block, something like 5 in. long, and in a small chamber just above it, and a similar computation shows that even if the specific volume of the steam was never greater than when saturated, it must have passed the thermo-couple, always within nine-tenths of a second, sometimes within a thirtieth of a second, and on the average within less than half a second of the time when the *first* of the superheating heat was put in. It would, therefore, have been extraordinary if Thomas had not found exactly the apparent swoop which he did, and I believe that this swoop is wholly due to moisture in suspension. Professor Reeve has come to the same conclusion by different reasoning in a recent number of Power.

7 If this contention that Thomas had wet steam in his saturation tests is justified, his data should evidently be recomputed, leaving the saturation tests out of account altogether, as was suggested by Reeve in Power. This makes unavailable several other tests, such as those from No. 80 on, there being nothing with which to compare them. The remaining 59 tests have been grouped in pairs in the usual way; thus, when there were four points, the first was taken with the third and the second with the fourth, and so in general. The results of such a recomputation are given in the following table.

8 This table shows two things. In the first place, none of the 31 values are near enough to the saturation line or far enough apart to give any information whatever about that line at any pressure, and the actual curve which Thomas draws becomes nothing more than a sheer guess, than which even Knoblauch's tremendous extrapolation—which is at least a rational guess—is preferable. In the second place practically all of the 31 values fit Knoblauch's curves within their limit of error. As the last column of Table 1 shows, Thomas' values average only about 2½ per cent higher than Knoblauch's curves. It should be noticed that the direction of this deviation is that which one would expect on the wet steam theory, for it is probable that not only the saturation tests but all those at low superheats would be affected by this source of error, the amount of wetness present diminishing with increasing superheat. It would follow that C_p would run too high, and this is just what it seems to do.

SUMMARY

9 (*a*) Thomas almost certainly had wet steam in his saturation tests and possibly in many of his other tests; (*b*) if his data are recom-

TABLE 1 COMPARISON OF RECOMPUTED THOMAS' VALUES OF Cp WITH KNOBLAUCH'S VALUES

Tests	Pressure	Saturated temperature Degrees fahr.	Average Temperature Degrees fahr.	*Cp* Thomas	*Cp* Knob-lauch	Difference Per cent
5–2	7	177	266	0.468	0.462	+1.3
6–3			302	0.470	0.463	+1.5
7–4			365	0.472	0.464	+1.7
12–9	20	228	318	0.478	0.472	+1.2
13–10			354	0.474	0.471	+0.6
14–11			417	0.476	0.471	+1.1
18–16	35	259	331	0.502	0.483	+3.9
19–17			367	0.474	0.480	−1.2
22–20			484	0.478	0.476	+0.4
27–24	55	287	377	0.506	0.493	+2.6
28–25			413	0.493	0.488	+1.0
29–26			476	0.482	0.483	−0.2
34–31	75	308	397	0.512	0.503	+1.8
35–32			433	0.508	0.495	+2.6
36–33			496	0.495	0.487	+1.6
42–40	115	338	446	0.527	0.510	+3.3
43–41			482	0.531	0.500	+6.2
44–42			545	0.491	0.489	+0.4
48–47	165	366	419	0.559	0.562	−0.5
51–49			509	0.535	0.504	+6.1
52–50			572	0.482	0.494	−2.4
57–54	215	388	478	0.550	0.532	+3.4
58–55			514	0.555	0.513	+8.2
59–56			576	0.512	0.500	+2.4
63–61	300	417	489	0.559	0.555	+0.7
64–62			525	0.554	0.521	+6.3
65–63			561	0.550	0.510	+7.8
75–72	500	467	522	0.607	0.624	−2.7
74–73			522	0.625	0.624	+0.2
79–77			675	0.531	0.513	+3.5
79–78			693	0.512	0.511	+0.2

puted in such a way as to minimize this source of error, the remaining points give no evidence whatever as to the shape of the saturation line, and very little as to the shape of any of the lines in the usual C_p

diagram; (*c*) on the other hand, regarded as isolated points, they agree surprisingly well with Knoblauch's extrapolated curves, even at the high pressures; and (*d*), their deviations from these curves are almost all in the direction which the wet steam theory predicts; therefore, (*e*) it is unwise, at the present time to attempt to replace Knoblauch's values of C_p as originally published by Thomas' values, or by any compromise based on Thomas' values; and (*f*), the only modification of Knoblauch's values that can be made with any certainty is that which Professor Heck suggests in the region of very high superheats.

Prof. C. C. Thomas A new series of experiments upon the specific heat of superheated steam is now in progress under the writer's direction at the University of Wisconsin, and it is expected that the results of these experiments will throw important light upon the question of the relative accuracy of the investigations already completed. The present experiments are being made by an entirely different method from that employed heretofore, and under conditions thought to be especially favorable to obtaining exceedingly accurate results. The writer hopes to be able to present these further results as soon as the experiments are completed and worked up.

The Author The useful end finally sought in a discussion of this subject must be, not the instantaneous specific heat c_p, but the quantity of heat h needed to raise the steam from the saturation temperature to some higher temperature t. Obviously, an exact location (thermally) of the zero or starting-point of this operation is essential. This starting-point of superheat is also the terminus of another most important operation, that of evaporation. Nothing has been made clearer by recent experiment than the fact that the state of dry saturation is exceedingly elusive, and very hard to produce or to maintain in a body of steam.

2 Professor Thomas used in his calorimeter a small current of steam, broken into very thin streams where it passed over the heating surface of the wire coils. While the action observed in the superheating section of the apparatus of Knoblauch and Jakob raises serious doubts as to whether Thomas really secured homogeneous dry steam to which to add his measured superheat, nevertheless the writer is much inclined to believe that Dr. Davis has over-estimated the magnitude of the error involved. It will be noted, however, that the final curves in Fig. 6 of the paper fully embody the idea that

Thomas was charging too much energy to the early part of the superheating process.

3 Even if the writer erred in following Thomas in making the curves of c_p on t sweep upward toward saturation, the amount of this error is small. As a concrete case, consider the first line of Table 1, for 1 lb. absolute pressure. In the accompanying tabulation, the first line t_s is degrees of superheat; the second is c, from the table; the third line c' is from an assumed curve which sweeps in to saturation without rise; Δh, in the fourth, is the difference in the heat h,

TABLE 1

t_s =	0.		10.		20.0		40.0		60.0		80.0	
c =	0.513		0.475		0.465		0.457		0.454		0.453	
c' =	0.450		0.450		0.451		0.451		0.452		0.452	
Δh =		0.44		0.20		0.20		0.08		0.03		

over one interval, due to the difference between c and c'. Adding up these values of Δh, we have a total of 0.95, or say one B.t.u. If chargeable to that cause, this would represent the effect of $\frac{1}{10}$ of 1 per cent of entrained moisture. Of course there is uncertainty here, and we need more precise information; but this small possible error in the superheat, measured from saturation, is certainly far less than the probable error in Regnault's value of the total heat, measured to that same "point" of saturation.

4 Knoblauch and Jakob made no attempt to go back to zero superheat experimentally. The extension of their curves to this limit is extrapolation, pure and simple, even though made with the best judgment. The writer cannot, however, accept the dictum that c_p must rise to infinity at the critical temperature, no matter what the mathematical deduction on which this idea is based. In the past much mental energy and ingenuity have been wasted in the effort to apply the general mathematical theory of thermodynamics to this subject of superheated steam, with an entirely insufficient basis of special and exact physical knowledge; nor has any such basis been established up to this time. The relation $c_p = \infty$ belongs to the evaporation process: it means that heat can be imparted under constant pressure without raising temperature. Consideration of the writer's Fig. 3 ought to convince anyone of the impossibility of such action taking place anywhere to the right of the saturation line *ECD*. If the initial c_p need not thus rise to infinity at 689 deg. fahr., it is perfectly permissible to decide that Knoblauch and Jakob may have

17, can meet and start at any other point than the origin of the system of coördinates used. What Thomas did was to find, as accurately as he could, the point where any further addition of heat to steam originally slightly moist just began to raise the temperature: then from this point as zero he began to measure the superheat h. Whether he succeeded in getting precise and entirely reliable results is another matter; but curves logically representing such experiments must come to the origin. It is not well to wrench experimental results into conformity with a preconceived rigid assumption, of which the especial merit is the possibility of expressing it by a simple mathematical formula. It would be very difficult to build a physical concept about the idea of some other "origin of superheat" than the zero-point.

6 The presentation of further experimental data in regard to superheated steam will be looked for with great interest: but what we now need is a thorough review of the properties of saturated steam. At present, our knowledge of the heat required above saturation is probably more accurate than what we know about saturated steam.

No. 1197

A RATIONAL METHOD OF CHECKING CONICAL PISTONS FOR STRESS

By Prof. George H. Shepard, Syracuse, N. Y.
Member of the Society

The following is the notation used:

A = the area of section, by an axial plane of the piston, of the volume under pressure; that is, in Fig. 1, A equals twice the area $b\ c\ e\ j$.

a = the area of the section of the material of the piston by a normal cone, at the point where the stress is calculated. In Fig. 1, $e\ h$ is the trace of the normal cone.

f = the stress, along the slant height of the cone, at any point. For example, in Fig. 1, f is the tensile stress, in the line $e\ c$, at e, or the parallel compressive stress at h.

M = the moment of resistance to bending of a section of the material of the piston by a normal cone.

p = the maximum unbalanced pressure on the piston per unit of area.

R = the resultant, per unit of circumference of radius r_2 (Fig. 1) of the resistance of the cone to spreading or collapsing.

r = the outside radius of the piston (Fig. 1).

r_1 = the radius from the axis of the piston to the neutral axis of a section of the material of the piston by a normal cone, at the point where the stress is to be calculated (Fig. 1).

r_2 = the radius of the cross-section of the volume under pressure by the plane of R (Fig. 1).

r_3 = the maximum radius of the conical part of the piston.

S = the total pressure on the piston, perpendicular to any axial plane. For example, in Fig. 1, S is the total pressure perpendicular to the plane whose trace is $O\,X$.

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

t = the thickness of the material of the piston at the point where the stress is to be calculated.

y = the radius of any point on the surface of the piston about the axis of the piston (Fig. 1).

θ = the angle between that axial plane of the piston, which is taken as the plane of reference, and any other axial plane (Fig. 1).

ρ_1 = the slant height of the cone, on the side under the larger pressure, at the point where the stress is to be calculated (Fig. 1).

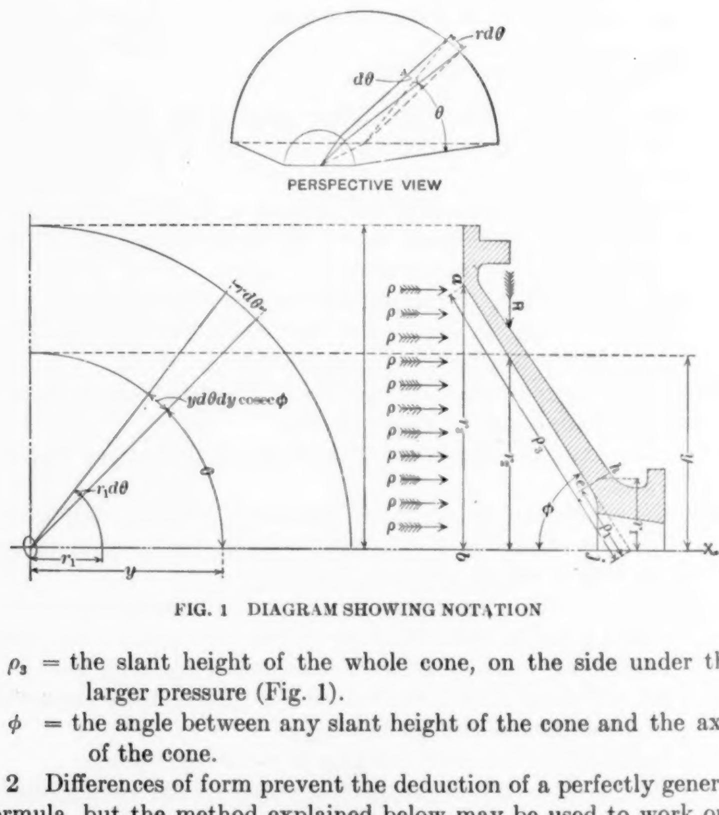


FIG. 1 DIAGRAM SHOWING NOTATION

ρ_3 = the slant height of the whole cone, on the side under the larger pressure (Fig. 1).

ϕ = the angle between any slant height of the cone and the axis of the cone.

2 Differences of form prevent the deduction of a perfectly general formula, but the method explained below may be used to work out the stress in any case.

3 Considering any frustum of the cone between the base and the surface of a normal cone whose trace is $e\,h$ (Fig. 1), it is apparent that it is acted upon by the following forces:

a The total unbalanced pressure, which is p times (area of surface of frustum, including the area of the circumferential flange).

b The resistance of the cone to spreading or collapsing.

The resultant of this acts in that plane, perpendicular to the axis of the cone, of which R in Fig. 1 is the trace, and is R per unit of circumference of radius r_2, the value of r_2 remaining to be found.

c The shearing force at slant height ρ_1, along the section by the normal cone.

d The tension and compression in the material, at slant height ρ_1 perpendicular to the section by the normal cone.

4 Consider as a free body an elemental sector, formed by two axial planes, making the angle $d\,\theta$ with each other (Fig. 1) and by the normal cone whose trace is $e\,h$, at slant height ρ_1.

5 First consider the piston as loaded on the inside of the cone, as as in Fig. 1.

Take moments about the neutral axis of the section $e\,h$.

Then for equilibrium

$$\text{(Moment of pressure)} = \left.\begin{array}{l}\text{(Moment of resistance of the cone to spreading)} + \\ \text{(Moment of resistance of cross-section } e\,h)\end{array}\right\}$$

6 Considering an elemental area on the surface of the cone,

Its area $= y\,d\,\theta\,d\,y\,cosec\,\phi$.

The pressure normal to the cone per unit of area $= p$.

Therefore the element of moment $= p\,cosec^2\,\phi\,(y - r_1)\,y\,d\,y\,d\,\theta$

and the moment of the pressure on the whole cone

$$= \int_{r_1}^{r_3}\int_0^{2\pi} p\,cosec^2\,\phi\;(y - r_1)\;y\,d\,y\,d\,\theta$$

$$= 2\pi p\,cosec^2\,\phi\left(\frac{r_3^3}{3} - \frac{r_1 r_3^2}{2} + \frac{r_1^3}{6}\right)$$

7 In addition to this there is the moment of the pressure on the circumferential flange

$$= p\int_{r_3}^{r}\int_0^{2\pi}(y - r_1)\;y\,d\,y\,d\,\theta$$

$$= 2\,\pi\,p\left(\frac{r^3}{3} - \frac{r_3^3}{3} - \frac{r_1 r^2}{2} + \frac{r_1 r_3^2}{2}\right)$$

8 Therefore the whole moment of the pressure

$$= 2\pi p \left\{ cosec^2 \phi \left(\frac{r_2^3}{3} - \frac{r_1 r_2^2}{2} + \frac{r_1^3}{6} \right) + \left(\frac{r^3}{3} - \frac{r_2^3}{3} - \frac{r_1 r^2}{2} + \frac{r_1 r_2^2}{2} \right) \right\}$$

9 The resistance of the cone to spreading, for the elemental sector,

$$= R\, r_2\, d\theta$$

10 The component of this, perpendicular to the reference plane, whose trace is OX (Fig. 1).

$$= R\, r_2 \sin\theta\, d\theta$$

11 If the cone be bisected by the plane of OX, the component perpendicular to OX, of the resistance of either half to spreading

$$= R\, r_2 \int_0^{\pi} \sin\theta\, d\theta$$

$$= 2\, R\, r_2$$

12 For equilibrium this must equal the total pressure, perpendicular to the axial plane whose trace is OX, which is equal to S.

Then $$S = p\,A$$

and $$2\, R\, r_2 = p\,A$$

Therefore $$R = \frac{p\,A}{2\,r_2}$$

13 Therefore the resistance of the whole cone to spreading, which equals

$$2\pi r_2\, R = \pi\, p\, A$$

14 Considering any elemental area on the surface of the cone, it is apparent that, if it is in a condition of equilibrium, the unbalanced pressure upon it in a direction perpendicular to the axis of the cone, must be directly and exactly counteracted by that element of the resistance of the cone to spreading which exists over that area; for, if this were not the case, there would result an immediate deformation of the elemental surface in the direction of yielding to the greater of these two forces; but this is inconsistent with the assumed equilibrium.

15 Therefore, the resultant of the resistance to spreading acts directly opposite to the resultant of the pressure perpendicular to the axis of the cone; and, since the pressure per unit of area is uniform, its resultant acts at the center of gravity of the area exposed to pressure.

16 Since the exposed area of the elemental sector is a trapezoid, its center of gravity is at a distance from the neutral axis of $e\,h$ of

$$\frac{2}{3} \cdot \frac{\rho_2^3 - \rho_1^3}{\rho_2^2 - \rho_1^2} - \rho_1 \text{ (very approximately)}$$

Therefore the arm of R (Fig. 1) about the neutral axis of $e\,h$

$$= \left(\frac{2}{3} \cdot \frac{\rho_2^3 - \rho_1^3}{\rho_2^2 - \rho_1^2} - \rho_1 \right) \cos\phi$$

17 Therefore the moment about the neutral axis of $e\,h$, of the resistance of the whole cone to spreading

$$= \pi\,p\,A \left(\frac{2}{3} \cdot \frac{\rho_2^3 - \rho_1^3}{\rho_2^2 - \rho_1^2} - \rho_1 \right) \cos\phi$$

$$= \pi\,p\,A \left(\frac{2}{3} \cdot \frac{\rho_2^2 + \rho_2\rho_1 + \rho_1^2}{\rho_2 + \rho_1} - \rho_1 \right) \cos\phi$$

18 Considering the resistance of the cross section at $e\,h$ to bending,

$$\frac{2.f}{t} = \frac{12.M}{at^2} \text{ (very approximately)}$$

and

$$M = \frac{\pi\,f\,r_1\,t^2}{3}$$

19 Substituting in equation [1],

$$2\,\pi\,p \left\{ cosec^2\,\phi \left(\frac{r_2^3}{3} - \frac{r_1 r_2^2}{2} + \frac{r_1^3}{6} \right) + \left(\frac{r^3}{3} - \frac{r_2^3}{3} - \frac{r_1 r^2}{2} + \frac{r_1 r_2^2}{2} \right) \right\}$$
$$= \pi\,p\,A \left(\frac{2}{3} \cdot \frac{\rho_2^2 + \rho_2\rho_1 + \rho_1^2}{\rho_2 + \rho_1} - \rho_1 \right) \cos\phi + \frac{\pi}{3}\,f\,r_1\,t^2$$

$$\therefore\ f = \left\{ 2\,cosec^2\,\phi \left(\frac{r_2^3}{3} - \frac{r_1 r_2^2}{2} + \frac{r_1^3}{6} \right) + 2 \left(\frac{r^3}{3} - \frac{r_2^3}{3} - \frac{r_1 r^2}{2} + \frac{r_1 r_2^2}{2} \right) \right.$$
$$\left. - A \left(\frac{2}{3} \cdot \frac{\rho_2^2 + \rho_2\rho_1 + \rho_1^2}{\rho_2 + \rho_1} - \rho_1 \right) \cos\phi \right\} \frac{3.p}{r_1 t^2} \qquad [3]$$

20 Equation [3] is too complicated to permit of direct design for dimensions; but the piston may be laid out by trial, guiding oneself by empirical formulae, and the stress at any point can then be calculated by equation [3]. It is to be remembered that, if the interior

TABLE I CALCULATED STRESSES FOR CONICAL PISTONS

Cylinder	Maximum unbalanced pressure. Pounds per square inch	Diameter of piston. Inches	Location at which stress is calculated	Angle of cone. Degrees.	Stress. Pounds per square inch	Material	Ultimate tensile strength of material. Pounds per square inch	Factor of safety on ultimate tensile strength	Tensile strength of material at elastic limit. Pounds per square inch	Factor of safety in elastic limit	REMARKS
H. P.	150.0	22.94	Just outside hub	91	9 300	Cast iron	29 000	3.12	......	....	Thickness of piston body is uniform. Stress shown is therefore the maximum existing
I. P.	59.0	34.0	Just outside hub	139	38 300	Forged steel Class A, No. 1	95 000	2.48	65 000	1.7	Maximum stress
I. P.	59.0	34.0	Quarterway down	139	18 550		95 000	5.12	65 000	3.51	
I. P.	59.0	34.0	Halfway down	139	12 800		95 000	7.43	65 000	5.18	
L. P.	28.3	37.0	Just outside hub	144	20 000	Cast steel	65 000	3.25	30 000	1.5	Maximum stress
L. P.	28.3	37.0	Quarterway down cone	144	12 500		65 000	5.2	30 000	2.4	
L. P.	28.3	37.0	Halfway down cone	144	7 270		65 000	8.95	30 000	4.13	

The above stresses were calculated for the conical pistons of the U. S. torpedo-boat destroyer Truxtun.

form of the piston is different from that shown in Fig. 1, the formula for stress must be deduced for the actual case by the method explained above. Ordinarily the maximum stress will occur just outside the hub.

21 It appears that the only quantity except t on the right hand side of equation [3] that is affected by changing t is r_1, and this is but slightly affected, so that f varies inversely as t^2, very nearly. If, therefore, calculation shows an unsatisfactory value of f at any point, t may be varied inversely as the square root of f within wide limits, in order to bring the stress to a proper value.

22 By the above method the stress, with pressure inside, was calculated for the conical pistons of the United States torpedo-boat destroyer, Truxtun, with the results shown in the accompanying table. The odd decimals in dimension result from scaling as nearly as possible from drawings.

23 The thanks of the writer are due to Prof. C. C. Thomas of Cornell University for the use of data and for other kind assistance in the preparation of Table 1.

24 Notwithstanding the differences of form, material, and unbalanced pressures, of the three pistons checked, they show factors of safety that agree fairly well at corresponding points. The minimum factor of safety shown in each case is also about what would be expected in this kind of work, in which every effort is made to reduce weight to a minimum.

25 In the above discussion the pressure has been taken as acting within the piston. When the pressure is on the outside of the piston, it may fail by buckling as well as in the ways considered above; but, in any practical case, the ratio of depth to thickness of piston will be so small that it is very probable that the piston will be stronger against buckling than against the ways of failure above discussed and therefore that only the latter need be considered. Variations in the form of the piston will affect the result and prevent the deduction of a general formula; but the application of the analysis above employed will enable a formula to be deduced for any particular case.

26 Strictly there is, in the case of inside pressure, a uniformly distributed compression; and, in the case of outside pressure, a uniformly distributed tension, over the section by the normal cone, which should be calculated and used to modify the result by the method above discussed; but this requires no explanation. In most cases this uniformly distributed stress may probably be neglected without serious error.

DISCUSSION

Mr. M. Nusim[1] Professor Shepard endeavors to arrive at a rational solution of the problem of the stress distribution in a conical piston without the aid of the theory of elasticity and by means of the principles of statics alone. He considers the equilibrium of an element of the conical shaped piston cut out by two infinitely near meridian planes. In arriving at the equation of equilibrium of such an element, the couple or bending moment acting on any meridian section of the piston is not taken into account.

2 The state of stress in a conical shaped piston subject to fluid pressure can be specified as follows: Consider an element of the piston (see Fig. 1) cut by two infinitely near planes through the axis of the

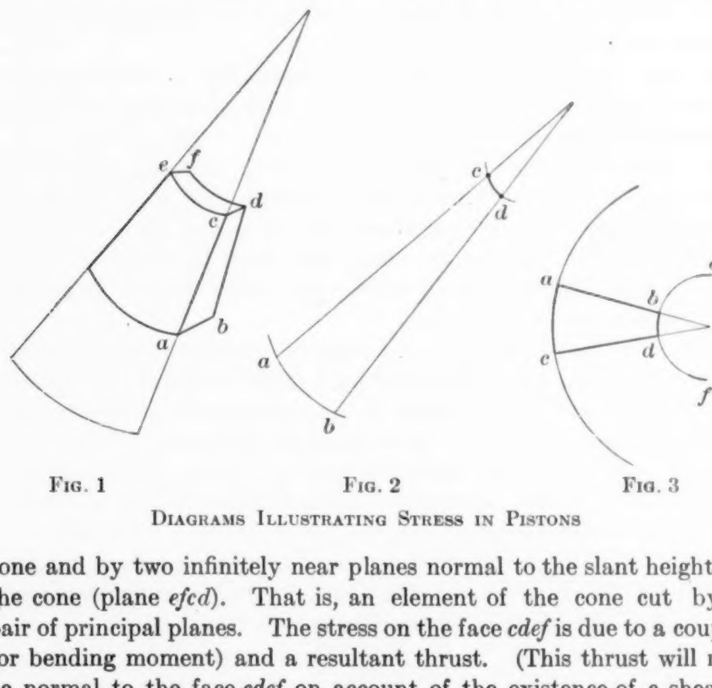


Fig. 1 Fig. 2 Fig. 3

Diagrams Illustrating Stress in Pistons

cone and by two infinitely near planes normal to the slant height of the cone (plane *efcd*). That is, an element of the cone cut by a pair of principal planes. The stress on the face *cdef* is due to a couple (or bending moment) and a resultant thrust. (This thrust will not be normal to the face *cdef* on account of the existence of a shear). Similarly, the state of stress on the face *abcd* can be specified by a couple and a resultant thrust normal to this face. In general both the couple and thrust on this meridian section will vary as we go along the slant height. The resultant thrust, over the whole section

[1] Mr. M. Nusim, General Electric Co., Schenectady, N. Y.

made by a plane through the axis of the cone, is of course given by the product of the fluid pressure and projected area of piston subject to this pressure.

3 If now we consider, as does Professor Shepard, a whole element of the piston cut by two infinitely near meridian planes (see Fig. 2), and desire to find the stress on the face whose trace is *cd*, say, we should take into account besides, the fluid pressure over the element *abcd*.

a The resultant thrust on the part of the meridian section between *b* and *d*.

b The resultant couple on the same part of the meridian section; for this couple has a component acting on the face whose trace is *cd*. It is this couple (bending moment) that Professor Shepard leaves entirely out of account in writing down the equation of equilibrium.

4 To bring out this point more clearly, consider a flat piston, Fig. 3. In this case the resultant thrust on any meridian section (or what Professor Shepard calls the resistance to collapse) vanishes, while the resultant couple over the same section has a definite value.

5 Suppose we desire to find the bending moment in the section whose trace is *bd*. It is evident that the couple on the faces whose traces are *cd* and *ab* have a component about *bd* which cannot be neglected. To neglect these couples (on the meridian sections) would mean to consider instead of a solid disc, one that is cut by a series of radial planes up to the circle *ebdf*.

6 The rational solution of the stress distribution in a conical shaped piston cannot be obtained by means of the principles of statics alone. One must have recourse to the mathematical theory of elasticity, otherwise one is apt to arrive at an approximation only.

THE AUTHOR Mr. Nusim's discussion shows that a conical piston is somewhat stronger than the stress would indicate, when calculated as explained in my paper. Mr. Nusim does not attempt any quantitative evaluation of this additional strength; but the stresses shown by my method for actual pistons in successful use (see Table 1) render it probable that the method gives a fairly close approximation to the true value of the stress. Mr. Nusim shows that a designer would at least be on the safe side in using the method. The most that the author would claim is that it gives the designer a fairly convenient means of assuring himself of the safety of any conical piston, without being obliged to rely upon empirical methods.

No. 1198

A JOURNAL FRICTION MEASURING MACHINE

By Henry Hess, Philadelphia, Pa.
Member of the Society

THE PURPOSE OF THE MACHINE

This machine was devised primarily to try out ball bearings by running them under various loads and speeds; the loads being applied either radially (normal to the shaft axis), axially (parallel to the shaft axis-thrust), or both simultaneously.

2 As it was desirable to note the behavior of the bearing throughout a test, some measure of its quality was necessary. A ball bearing or, for that matter, any bearing, must be capable of carrying its load without failure, and incidentally should waste the smallest possible energy in friction.

3 The measures of bearing quality would therefore be *a*, carrying capacity, and *b*, frictional resistance.

4 This machine is capable of applying radial loads up to a maximum of 15 000 pounds, and thrust loads up to a maximum of 10 500 pounds under rotative speeds ranging from 150 to 2500 revolutions per minute. It is capable of measuring the coefficient of friction when radial load only is applied, when thrust load only is applied, and when both radial and thrust load in any relative amount are applied.

5 The scheme of the machine is very simple, as is apparent from the skeleton diagrams, Fig. 1 and 2.

6 The bearing A to be tested is mounted at the forward end of a shaft that is rotated by a reversible motor. The bearing box is held in an embracing strap AB which is hinged to the rod BC. The upper end of rod BC is connected to a system of weighing levers typified in the diagrams by a single beam CE with its weight Q.

7 So long as the bearing is at rest, the line from the center of the bearing A to the hinge of the strap, and from this hinge to the weighing beam attachment C will form a continuous straight line AC.

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

Rotating the bearing in the direction of the arrow will deflect the strap AB and rod BC as shown in Fig. 2. The journal friction is the deflecting force and is balanced by the weight Q acting through the weighing beam. The deflection of AB and BC from the straight line is a measure of the deflecting force; that is, of the journal friction.

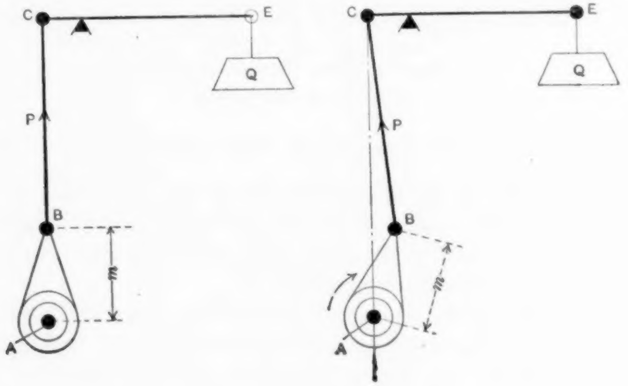


Fig. 1 Fig. 2

Diagrams of Radial Friction Measuring Arrangement

8 The arrangement for applying thrust load is shown in skeleton diagram in Fig. 3. The bearing A is carried as before at the end of a shaft, through whose bore a wire G is passed; the forward end of this

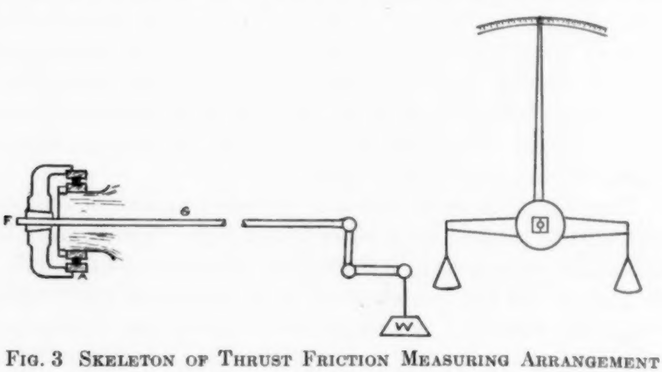


Fig. 3 Skeleton of Thrust Friction Measuring Arrangement

wire carries a yoke F that rests against the outer race of the bearing so that the load pulling on the wire through a system of weighing levers puts a thrust load on the bearing. A delicate balance with pointer and weight pans is attached to the yoke. The journal friction

tends to rotate the yoke and twist the wire until the latter's torsional resistance balances the friction. Restoring the balance by weights in the high pan measures the friction directly.

9 When the journal is submitted to combined radial and thrust

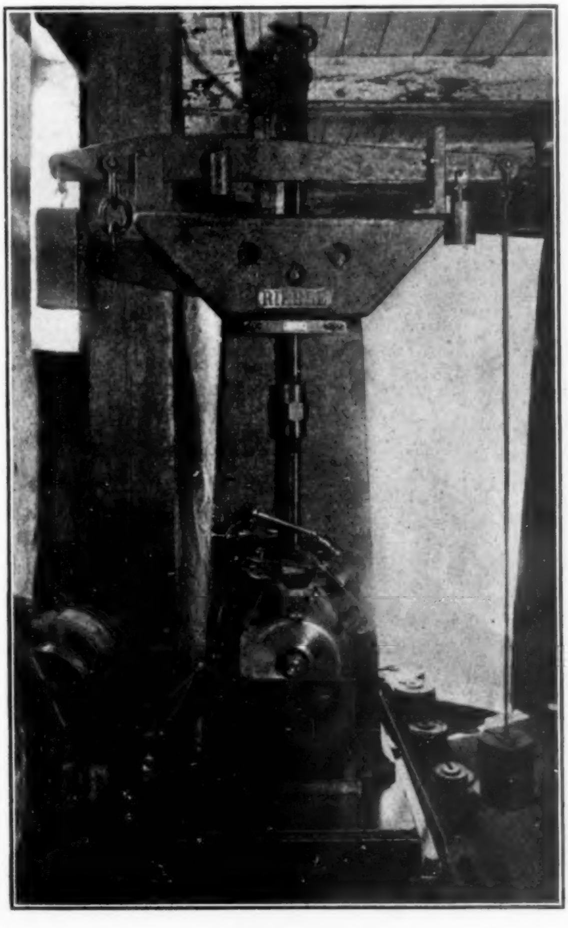


FIG. 4 MACHINE ARRANGED TO MEASURE THE FRICTION OF A RADIALLY LOADED BEARING

loads the radial load is applied as above by the weight Q, Fig. 1, acting through the rod BC; and the thrust load by the weight W, Fig. 3, acting through the wire G. The total friction due to the combined

load is then measured by determining the deflection of the strap *AB* and rod *BC* of Fig. 1. As will be explained later, a correction is made for the torsional resistance of the thrust-applying wire, as this affects slightly the amount of deflection of *AB* and *BC*.

THE ACTUAL MACHINE

10 Fig. 4 is a photographic view showing the machine arranged for

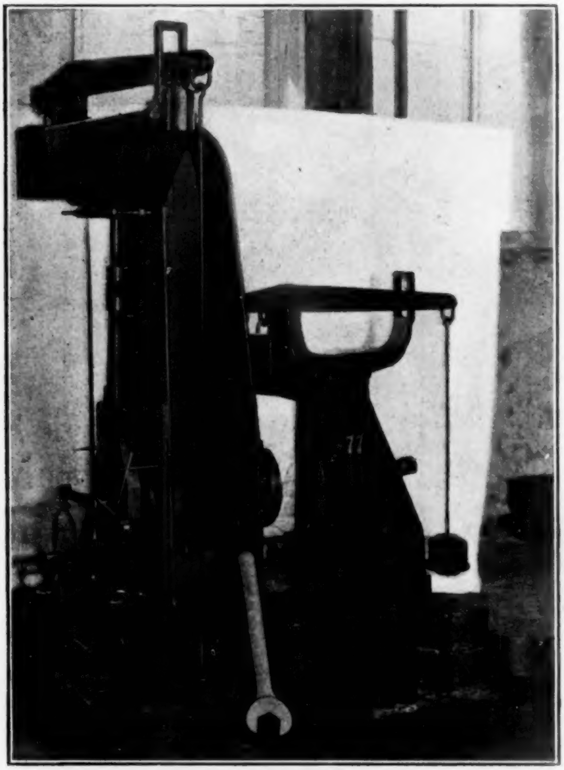


FIG. 5 MACHINE ARRANGED TO MEASURE THE FRICTION OF A THRUST LOADED BEARING

testing a bearing under radial load only. Fig. 5 shows it for thrust load only. Fig. 6 shows front and side views of the machine, in which the arrangement of the weighing levers is indicated.

11 The machine has a spindle, Fig. 7, of large size compared with the loads to be carried, mounted on heavy ball bearings and driven by

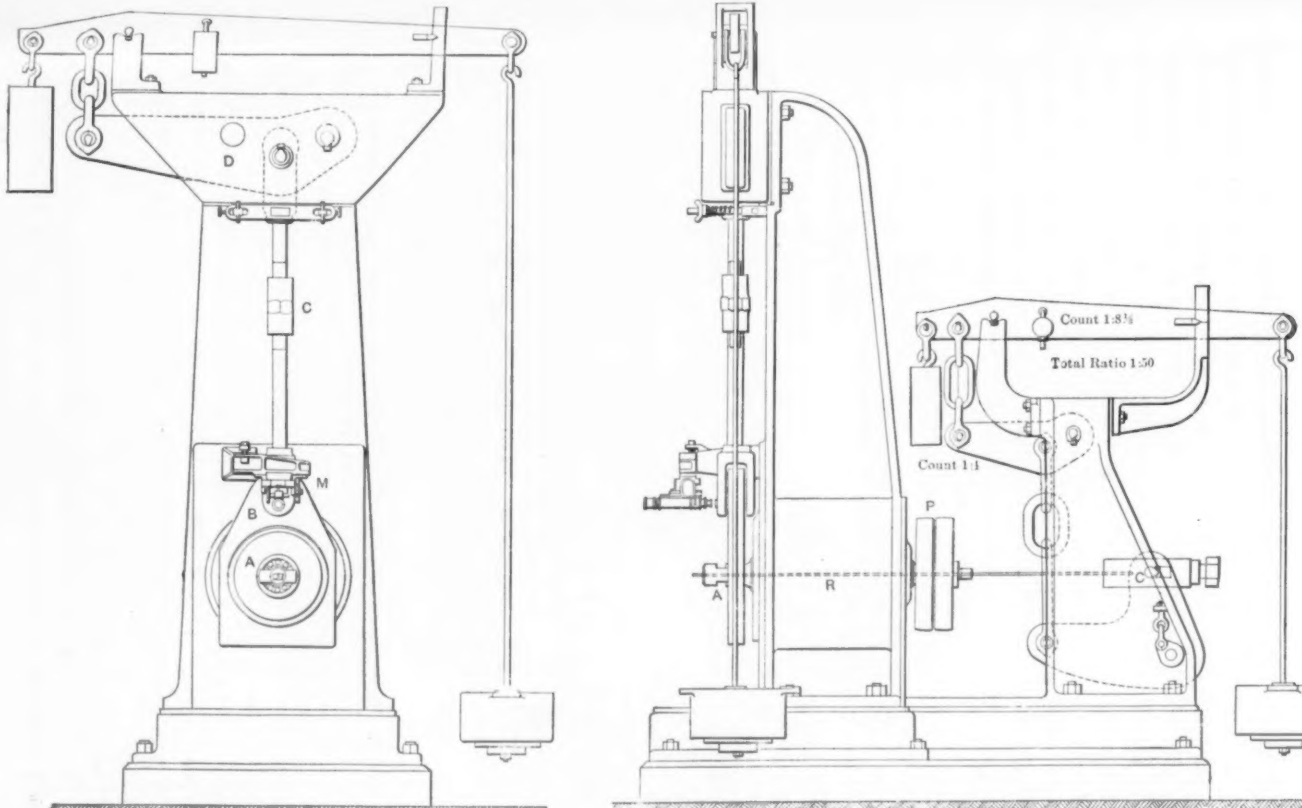


FIG. 6 FRONT AND SIDE ELEVATION

A, BEARING TO BE TESTED; P, DRIVING PULLEYS; B, STRAP HOLDING OUTER RACE OF TEST BEARING; M, MICROSCOPE TO MEASURE DEFLECTION; C, ROD CONNECTION STRAP WITH WEIGHING LEVER D; R, THRUST APPLYING WIRE ATTACHED TO WEIGHING LEVER C; WEIGHT PANS ATTACHED TO A YOKE AT A' FOR MEASURING TORSION, NOT HERE SHOWN

belt from an interpolar variable speed motor whose range is increased by an exchange of pulleys. The front end of the spindle is bored out to receive a taper arbor of which a number are provided to suit the various bearings to be tested. The inner race of the test bearing is mounted on the overhanging nose of this taper arbor, while the outer race is carried in the yoke suspended from the rod connected to the loading beams. The rod itself is made of two pieces connected by a differentially threaded turnbuckle. The system of weighing levers, clearly shown in the illustrations, pulls upward on the bearing and so loads it. With the bearing in place and the proper weights hung, the

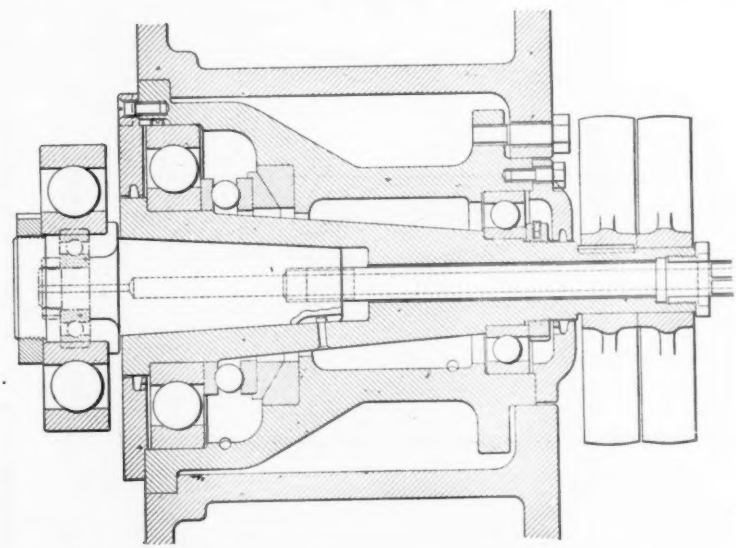


Fig 7 Section of Spindle; Bearings to be Tested are Mounted on Arbor at Left

turnbuckle is adjusted until the index in the right-hand end of the long top-beam lines up with the mark showing that all is in proper balance. The bearings which support the spindle are sufficient to allow radial loads of 15 000 pounds and thrust loads of 10 500 pounds. The regular working of the machine entails a severe trying out of these bearings. The combination of a load of 15 000 pounds and the high speeds employed is particularly trying.

12 A microscope with focal and transverse adjustments is horizontally mounted to observe the deflection of the bearing yoke and load

rod. It will read ten thousandths of an inch on a fairly open scale ($\frac{1}{32}$ inch divisions), so that interpolations are quite possible. The crosshairs in the eyepiece are brought over the line on the rod end with the bearing at rest; slight vibration by hand ensures the correct alignment of the bearing center, yoke pivot and upper rod pivot. While running, the microscope is traversed horizontally until crosshairs and reading line again coincide. In practice it is found that at certain speeds the vibrations of the framing and of the loading system coincide in phase occasionally so that the reading line swings rather far to either side; these oscillations die down as the phases fall out of step so that sufficiently definite readings are secured. The reading line is a very fine scratch on a highly polished silver plate. Direct illumination from an incandescent lamp has given the best results—better than light reflected through the objective by a prism in the microscope barrel.

13 The thrust load is applied to the bearing under test by a yoke that rests against the outer race or box. This yoke takes a wire rod by taper clamps. The wire passes through the hollow spindle to the upper arm of a bell crank weighing lever, being clamped to that by similar taper jaws. A tension applying screw is carried by this lever. The entire thrust load applying scale is, with the exception of the first bell crank beam, of standard construction and conveniently arranged on the back of the base.

14 The photographic view, Fig. 5, shows quite clearly the arrangement of the weighing beam that is attached to the loading yoke, together with its weight pans and long indicating needle. The scale responds sensibly to weights of 0.0005 pound placed in the pans.

PRECAUTIONS OF DESIGN

15 Since the coefficient of friction of a good ball bearing is as low as 0.0015 referred to the shaft diameter, it follows that errors of the machine may be but small. The application of loads up to 15 000 pounds demands a decidedly robust construction. The general scheme of the radial load friction apparatus is in principle modeled on a design of Professor Stribeck's, but differs from his in the use of rod *BC* Fig. 1, as a tension member instead of a compression strut as he had it. Obviously, a strut of any length (36 inches in this machine) to transmit 15 000 pounds would have to be very heavy and cumbersome.

16 In the Stribeck machine the deflection for AB and BC from the straight line is measured by a system of multiplying levers; these necessarily involve frictional resistances, which, though slight in themselves, are nevertheless undesirable. A direct reading of the deflection itself by means of a microscope reading to one ten-thousandth of an inch is substituted.

17 The various pivots are all knife edges. The beams and rods themselves were carefully calculated to balance and then corrected in the actual building. This assured the essential high degree of sensitiveness.

18 Attention may here be drawn to the fact that the frictional resistance of the shaft carrying the bearing to be tested does not affect the result in any way. It would otherwise be obviously impossible to measure the friction of a bearing taking on only a few hundred pounds load, and in the same machine, another taking fifteen thousand pounds. As the shaft-supporting bearings must be capable of safely carrying this high load, their frictional resistance would otherwise totally overlay and obscure the resistance of the smaller bearings under test.

19 The point C, Fig. 1, should be vertically over A since the deflection of AB and CB is measured horizontally. As there is a possibility that C may not occupy this absolutely correct position and as an error to one side or other would increase or decrease the apparent deflection reading, the bearing is run in both directions for each test; the arithmetical mean of the two readings gives the correct one.

MAIN PROPORTIONS

20 By consultation with Riehle Brothers Testing Machine Company, the firm which built the machine to the author's design and specifications, a total lever ratio of 1 : 50 was fixed for both the radial and thrust load applying systems.

21 In Fig. 1 let m be the distance from A to BC—normally to BC, r the radius of the bearing at the bore or shaft surface and μ the coefficient of friction referred to radius r. Let P equal the load acting along BC.

There will be equilibrium when

$$Pm = P\mu r$$

$$m = \mu r \qquad [1]$$

m is therefore directly proportional to the frictional resistance of the

bearing under test. The shaft radius is chosen as the lever arm to which the friction is referred, as the friction acts at that lever arm in a plain journal. In a ball bearing or roller bearing the friction really acts at the tracks on which the balls roll, but for purposes of comparison with plain bearings it is referred to the bearing bore, which is identical with the shaft diameter.

22 The determination of the friction by direct reading of the deflection m was not quite convenient from the mechanical arrangement of the parts. A line was therefore scratched at a point lower

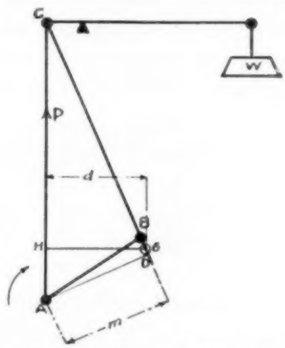


Fig. 8 Skeleton of Radial Friction Measuring Arrangement

down on the rod BC at G (see Fig. 8). The various dimensions chosen were

$$AC = 45 \text{ inches}$$
$$BC = 36 \text{ inches}$$
$$AB = 9 \text{ inches}$$
$$CG = 37.5 \text{ inches.}$$

This gives

$$AC : CG = 6 : 5$$

so that

$$\frac{AC}{CG} = \frac{6}{5}$$

23 It is clear from Fig. 8 that

$$\frac{AD}{AC} = \frac{HG}{CG}$$

and substituting m for AD, d for HG, $\frac{6}{5}$ for $\frac{AC}{CG}$ gives

$$m = \frac{6}{5}\, d$$

As it was found that $\mu = \frac{m}{r}$

then

$$\mu = \frac{6}{5}\frac{d}{r} \qquad [2]$$

Equation 2 is used for determining the coefficient of friction μ from the reading of the observed linear deflection of point G. As thrust loads are applied up to a maximum of 10 500 pounds and the friction is determined by weighing the deflecting force, it follows that a wire heavy enough to apply the maximum load would be deflected so little under the lighter loads as to demand extremely sensitive weighing apparatus. Several wires of different diameters are therefore used to cover the range.

24 When determining the friction due to simultaneous radial and thrust loading the microscope reading method, as used for the radial loading alone, is employed. But the torsional resistance of the thrust applying wire opposes the bearing friction as a deflecting force and so injects an error into the observation. This error is to be kept down to an amount which is negligible, say 0.5 per cent, by the suitable relative proportioning of radial load, thrust load and torsion wire; these are determined as follows:

25 The "*Taschenbuch des Vereins Hütte*" gives for straight torsional springs:

$$M_t = \frac{d\ \pi\ w^4\ G}{32\ t\ (AG)}$$

M_t = torsion moment
w = wire diameter
G = modulus of shear = 11 000 000 pounds per square inch
t = length of wire = 40 inches
d = arc through which wire is twisted
AG is that distance in Fig. 8.

As the angular deflection of the wire is very small, the arc may be considered a straight line without introducing a material error.

Inserting values gives

$$M_t = \frac{d\ \pi\ w^4\ 11\ 000\ 000}{32 \times 7.5 \times 40} = 3600\ d\ w^4$$

In the apparatus, Fig. 8, the frictional moment

$$M_f = P\ m = \frac{P\ 6\ d}{5}$$

26 The percentile relationship of the torsion and friction moments is defined by

$$E = \frac{M_t}{M_f} 100, \text{ and by substitution}$$

$$E = \frac{3600 \ d \ w^4}{P \frac{6}{5} d} 100 = 300\,000 \ \frac{w^4}{P}$$

27 For a desired error of 0.5 per cent we have

$$0.5 = 300\,000 \ \frac{w^4}{P} \text{ and}$$

$$P = 600\,000 \ w$$

28 A series of wires for applying thrust loads were specially drawn of high grade stock and the following table worked up for them.

TABLE I

Wire gage	Wire diam. in.	Max. thrust load lbs.	Min. radial load P for max. error of 0.5 per cent lbs.	Tensile stress in wire due to max. load lbs. per sq. in.
32 Music	0.086	1000	30	172 000
8 B & S	0.128	2000	160	155 00C
6 B & S	0.162	3000	400	145 000
4 B & S	0.204	5000	1000	155 000
2 B & S	0.258	8000	2600	181 000

29 Smaller radial loads than these given in the table can be used, but in that case the error becomes larger, so that readings must be taken by both the microscope and the torsion weighing scale; it was not thought worth while to incur this complication.

TYPICAL BEARING TESTS

30 As this is primarily a description of the friction testing machine and not of friction tests, conclusion will be made with a few typical tests that illustrate the range and accuracy of the machine.

31 Fig. 9 gives the results of a test of thrust bearing 9 *r* at 495 revolutions per minute under loads ranging from 500 to 1300 pounds. One point only, at 1000 pounds, falls outside of the range of a fairly smooth average curve, with the friction coefficient ranging from 0.075 to 0.080 of one per cent.

32 Fig. 10 gives similar curves for various speeds ranging from 265 to 655 revolutions per minute, while Fig. 11 gives a curve deduced

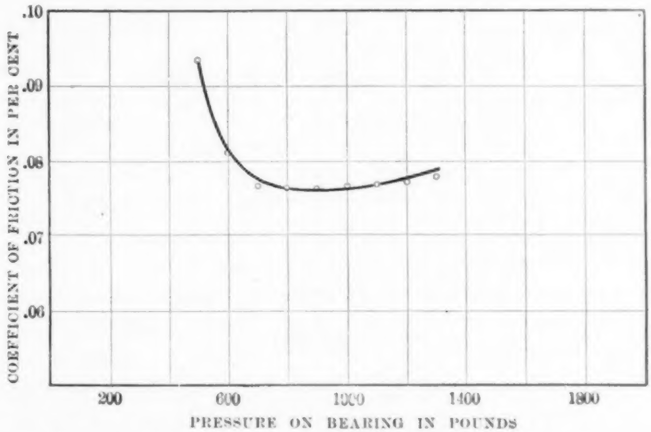


Fig. 9 Friction Characteristic of a Collar Thrust Ball Bearing at 495 r. p. m.

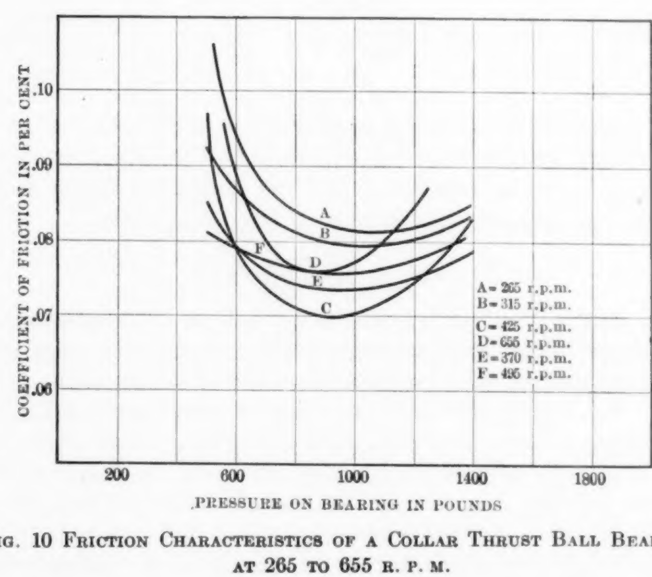


Fig. 10 Friction Characteristics of a Collar Thrust Ball Bearing at 265 to 655 r. p. m.

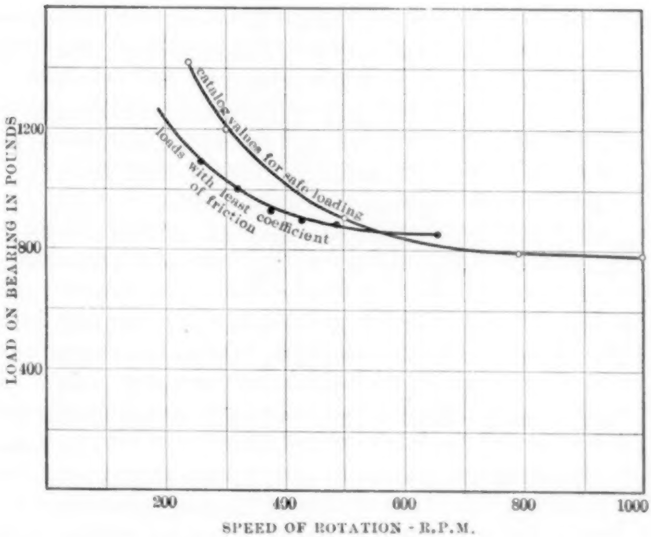


FIG. 11 LOAD AND SPEED CHARACTERISTIC OF A COLLAR THRUST BALL BEARING

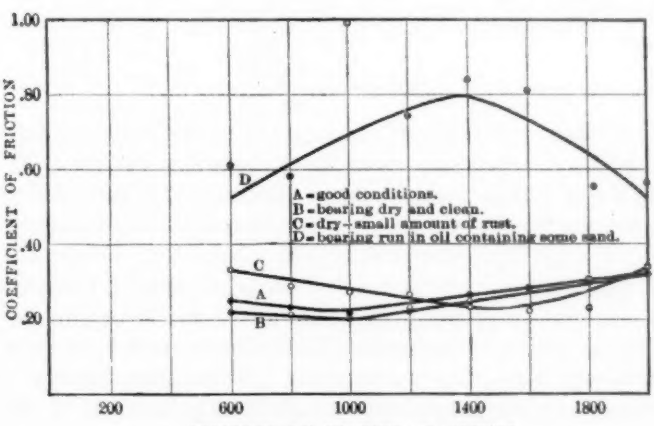


FIG. 12 FRICTION CHARACTERISTICS OF A RADIAL LOAD BALL BEARING UNDER NORMAL CONDITIONS AND UNDER ABUSE

from the observations and shows graphically the relationship between load capacity and speed, that load being considered the correct one for a certain speed at which the coefficient of friction is least.

33 The behavior of radial bearing No. 308 under radial loads only, as measured by the coefficient of friction, is plotted in Fig. 12. This was tested, first immersed in oil, then thoroughly washed clear of oil in gasolene and dried, then slightly rusted.

34 The rated load for steady load and uniform speed is 1400 pounds. The perfection of finish of the races and balls and truth of the parts is shown by the very small difference between the oiled and dry bearing. At about 25 per cent overload the influence of deformations in their effect on the true rolling became manifest in a crossing of the oil and dry curves, with the latter becoming higher. As was to be expected, the slightly rusted bearing showed the higher friction; but the unexpected result is shown of this friction becoming less than that of the unrusted bearing at about 1400 pounds; but between 1800 and 2000 pounds the friction rises sharply, indicating a probable early destruction of the bearing under prolonged running. Because dry and oiled curves show a similarly low friction, it is not safe to deduce the permissibility of running without oil. Prolonged dry running will surely prove destructive.

35 It is with pleasure that I acknowledge the efficient aid rendered by Mr. Messner in the detail design of the machine and Mr. Batt's careful and painstaking observations.

DISCUSSION

MR. J. ROYDEN PEIRCE[1] During the past two years, the writer has had an extensive experience in the practical use of ball bearings under trying circumstances and is glad to see a scientifically constructed machine to determine accurately the friction in such bearings.

2 Engineers in general are not so much interested in the differences in the internal frictional qualities of various ball bearings as they are in their reliability. The internal friction is too small to be an appreciable element in machine construction. On the other hand, an instrument such as the journal friction measuring machine of Mr. Hess should be of great value in detecting the inferiority of many of the

[1] J. Royden Peirce, President of The Royden Marble Machinery Co., 23rd St. and Madison Ave., New York.

ball bearings at present on the market. The final criterion of ball bearing quality is undoubtedly its frictional resistance.

3 If it is true that "Plain, simple words best pierce the ear of grief," it is doubly true that plain, simple bearings are the only ones which stand up satisfactorily under the rough duty of actual usage. It would be very interesting to have a set of tests from this machine of all the different bearings, less with a view to finding the actual friction than to finding the relative advantages or disadvantages of the various ball separating devices under different load and speed conditions.

4 That the friction in ball bearings is extremely small, will be seen from the following personal experience: On 207 ft. of 2 $\frac{15}{16}$ in. line shafting with 32 bearings, a light string passing over a 6 ft. pulley easily turned the entire stretch of shaft with its various pulleys, but without belts.

5 A marble rubbing bed weighing 16 tons and having a moment of inertia of 20 200, continued to run for 25 min. after the power was turned off.

6 By substituting in the well known formula

$$M = \frac{d^2 e}{I\ dt^2}$$

a value of the moment of retardation equal to 56 lb. is found. The inner diameter of the footstep thrust is $5\frac{1}{2}$ in., consequently

$$u = \frac{246}{32\ 000} = 0.0077$$

7 This includes the slight friction of two radial bearings in the stem and a couple of ordinary bearings on a horizontal quill running light. However, it is an example of regular practical operation and shows what an exceptionally small coefficient of friction ball bearings have. The bearings of these beds were remarkable as having been constructed for a weight of 4 tons instead of 16, and, with one or two exceptions, after eight months' duty they are still running in apparently perfect order.

8 The writer is using bearings on spindles running from 1500 to 2400 r.p.m. operating carborundum wheels. Some of these bear a weight of nearly a ton, which is an excessive load at this high speed, notwithstanding which they last six months with an eight hour day service and then require only a new set of balls. When the power

is thrown off, the speed decreases slowly and shows a good coefficient of friction.

9 Improperly designed cages or ball-separating framework may cause excessive wear and heating. An earlier experience showed that by the time overheating called attention to trouble, the bearing was ruined. The clearance and dimensions of the balls and raceways in a well constructed ball bearing are necessarily so exact that any distortion by heating either breaks the balls or ruins the raceways.

10 A testing machine that is sensitive enough to detect small changes in the frictional resistance of ball bearings should render valuable aid and engineers will eagerly await a complete set of experimental data from it.

Prof. John A. Brashear I would like to add a note as to the early use of ball bearings.

2 In or about the year 1890 we had occasion to take apart and repair an old 13-in. equatorial telescope belonging to the Allegheny Observatory, the mechanism of which was constructed by the Gurleys of Troy in 1856.

3 At the lower end of the polar axis we found a grooved steel ring with about a dozen $\frac{3}{8}$-in. hard bronze balls, some of which were in very good condition. I would like to know if any member recalls an earlier date for the use of ball bearings.

Prof. C. H. Benjamin It is evident that Mr. Hess in devising this machine had in mind a machine for measuring the slight friction of ball and roller bearings. Experiments I have made on this class of bearings have shown me that it is very difficult for the ordinary machines to work accurately enough, on account of the very small coefficient of friction with reference to the surface of the journal.

2 In order to do this several things must be considered. In the first place, the bearing to be tested should itself be insulated as completely as possible from any other bearings in the machine—so that the heat generated by the bearings of the driving shaft or the auxiliary shafts shall not be communicated to the bearing which is to be tested, and this would necessitate a different method of construction from that indicated.

3 According to my experience with journal friction machines, of which I have had quite a little in the last few years, most of the machines are too complicated, and I think there is still room for a simple machine with which commercial journals can be tested under

commercial conditions, one capable of carrying the heaviest loads which such journals would be required to bear, and having its test journal insulated as before mentioned. The temperature of the bearing is usually quite as important as the coefficient of friction.

4 With all due credit to the various manufacturers of journal machines now on the market there is still great room for improvement. This criticism, however, does not apply to the machine described in the paper because, as I said before, it is manifestly intended for testing ball bearings and not ordinary journals.

The Author Mr. Peirce's experience with ball bearings illustrates the fact, not yet generally recognized, that well made ball bearings are capable of abuse far beyond the endurance capacity of a sliding journal. Take for instance the footsteps of his rubbing beds; the arrangement usually employed is the hardened steel washer step running in oil and surrounded by a water cooling bath; but quite aside from the frictional losses, such footsteps are a source of more or less continuous trouble. A ball bearing, however, installed to carry 4 tons, is carrying over four times this load, and so successfully that the new properly dimensioned ball steps ordered when the error in the weight specifications was recognized have not yet been installed.

2 Mr. Peirce refers also to the ball bearings on his carborundum spindles. The exceedingly difficult character of this service will be recognized by every engineer familiar with the hammering effect of high speed parts not in perfect running balance. In this case we have a spindle carrying a series of carborundum discs 16 in. in diameter and combined to make up a cutting face 6 ft. in length, the discs strung on a cast iron spider. By the aid of balancing ways a good standing balance is secured, but a good running balance only when the fates are propitious. As an additional difficulty, ball bearings not being provided for in the original design, such dimensions were given them as the existing parts permitted, after a trial on plain journals had failed, with the expectation that frequent renewal would be necessary. Yet with all these disadvantages, renewal of the balls alone was necessary after six months continuous operation. This illustrates one of the decided advantages of the ball bearing. In the plain journal wear affects spindle and box, while in the ball bearing the balls and races are parts attached to the shaft, and their removal and replacement is all that is required to restore the journal to its pristine condition; no machining of any parts is required, but only reassemblage of relatively cheap parts.

3 Professor Brashear is interested in the early use of ball bearings. Their use goes far back of the Allegheny Observatory erection in 1856. The author has been informed that a cast iron sheave, supposed to have been installed over two centuries ago, is still to be seen on the quay at Dieppe, France. All records of its installation have been lost, and it has not been in use for over a century. The hub is broken, exposing to view part of the balls.

4 Whether a simpler machine for testing ordinary journals is desirable depends altogether upon the accuracy and refinement required. It is possible to determine the frictional qualities by actual measurement of the coefficient of friction on various existing machines of other types; since the friction of plain journals is relatively large, machines of lesser accuracy will answer for them. It must not be assumed, however, that the great delicacy of the machine described implies a lack of robustness—quite the reverse. Bearings have frequently been run to the point of total and violent breakdown at high speeds and under loads of 15 000 lb. without any damage whatever to the machine.

5 Professor Benjamin's demand that the insulation of the bearing to be tested from any other bearings in the machine be as complete as possible, is fully met so far as the resistance of these other bearings to friction is concerned. In that respect, isolation is absolute in this machine. Whether the frictional resistance of the spindle journal be high or low is absolutely immaterial. Ball bearings have been chosen simply because of their easy-running qualities and greater reliability under the extreme speed and load; manifestly it would be difficult to run a spindle 5.5 in. in diameter at 2500 r.p.m. under 15 000 lb. load in a plain journal.

6 Heat isolation, though not quite absolute, is also very good. For very accurate work the spindle is run at proper speed and load until the temperature of the machine journal is constant before the test bearing is put in place. The lower portion is submerged in a pan of oil, of which it carries a portion around with it, throwing it off in a narrow stream. A sensitive thermometer is placed in the path of this stream. While the temperature of this stream is not absolutely that of the journal, it is very nearly so, quite nearly enough for purposes of comparison; the method is as accurate as any with which the author is familiar.

7 When the endurance of a bearing is to be tested, a much simpler machine can be extemporized from wooden beams, plain shafting

and hangers, with a very simple loading lever, and answering every purpose.

8 The occasion of the design of the machine described was the observation of journal phenomena quite beyond the capabilities of the usual friction measuring machine, without limiting its load capacity, but rather while carrying that to a higher point than usual. While the author has so far confined the use of this machine to an investigation of ball and roller bearings, being primarily interested in carrying these to the highest attainable stage of perfection, he does contemplate a later investigation of plain journals, and hopes, because of the greater refinement and accuracy of this machine, to find the cause for certain phenomena in sliding journals and lubricants for which the usual apparatus fails to account.

No. 1199

SOME PITOT TUBE STUDIES

THE DISTRIBUTION OF VELOCITIES AND PRESSURES IN STRAIGHT AND CURVED PORTIONS OF A SIX-INCH WATER PIPE

By Prof. W. B. Gregory, New Orleans, La.
Member of the Society

and Prof E. W. Schoder,[1] Ithaca, N. Y.
Non-Member

It is the nature of science that general effects should be the first to be observed and studied. Later there come the more and more detailed investigations that make for an exact science. To this manner of evolution hydraulics is no exception.

2 In the design of water wheels, pumps, air blowers and gas engines, the improvements have come as the result of numberless tests involving accurate measurements of total quantities of discharge or supply and therefore of mean velocity of the liquid or gas. Such measurements may be *direct* as in the case of tank measurements, of total discharge in a given time, or indirect as in the case of measurement by a Venturi meter or Pitot tube. However, if we desire a knowledge of the *distribution* of velocities and pressures in a pipe or passage as well as of the total flow, then the Pitot tube is the only instrument available.

3 The writers have made a study of the distribution of velocities in a six-inch pipe by means of the Pitot tube, undertaken in connection with experiments on the loss of head due to elbows of varying radius. However, this paper will not discuss the loss of head in elbows. It was desired to know how far down the pipe the distortion in velocities due to the presence of the curve extended, or more exactly, to know whether there was elliptical distribution of velocities at distances of approximately 168 diameters and 76 diameters beyond the curve.

[1] In charge of Hydraulic Laboratory, Cornell University.

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

4 The distribution of velocities in the curve was also investigated. Traverses were made with a Pitot tube both in the plane of the curve and across the pipe at right angles with this plane. And a method was developed whereby the pressure at any point in a pipe, whether straight or curved, may be obtained by means of the Pitot tube.

5 The arrangement of the pipe line is clearly shown in Fig. 1 and Fig. 2. The pipe line was set up in Fall Creek Gorge, near the Cornell Hydro-Electric Power Plant, so that the water supply could be taken

FIG. 1 VIEW OF CORNELL UNIVERSITY HYDRO-ELECTRIC POWER PLANT, SHOWING SIX-INCH PIPE LINE USED IN THE PITOT TUBE EXPERIMENTS

from the blow-off end of the five-foot steel supply pipe where a head of approximately 135 ft. is available.

6 The curve in which the investigations with the Pitot tube were made was a 90 deg. circular elbow with radius of center line 2.5 ft., or five diameters. It was bent from six-inch wrought iron pipe. The ends were threaded for ordinary flanges.

7 The Pitot tube used is shown in place in Fig. 3 and Fig. 4. The details of the tube are shown in Fig. 6.

8 Traverses were made in both horizontal and vertical planes at points on the pipe 1.10 ft. upstream from the curve, at approximately 22½, 45 and 67½ deg. in the curve and also at points in the straight

pipe beyond the curve distant 84.10 ft. and 38.0 ft., respectively. See Fig. 2, 3 and 4.

9 In every case where a traverse was made in straight portions of the pipe line the static pressure at the wall of the pipe was obtained from two diametrically opposite openings in the same cross section as the Pitot tube and at right angles to the diameter on which the traverse was made. These holes were tapped out to receive one-eighth inch brass cocks. After drilling and tapping each hole the burr was

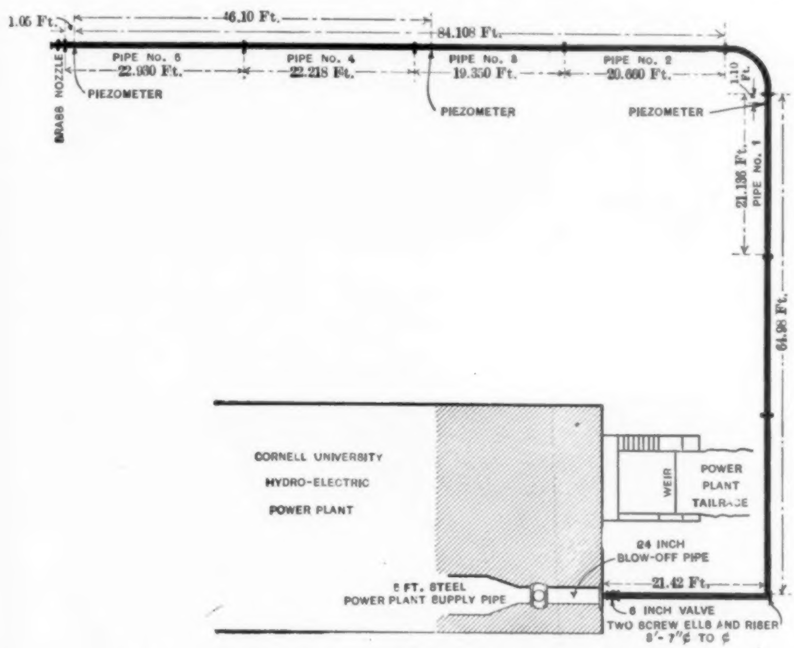


Fig. 2 Plan of Six-inch Wrought Iron Pipe Line

carefully removed from the inside of the pipe with a file, and care was taken that the cocks should not extend beyond the inner wall of the pipe when screwed home.

10 For all traverses in the curve the static pressures were obtained from the two horizontal openings 1.10 ft. upstream from the curve. In making traverses, readings were obtained at one-half inch intervals in some cases, while in others the points were so distributed that the arithmetical mean could be used directly in getting mean velocity. This matter will be discussed presently.

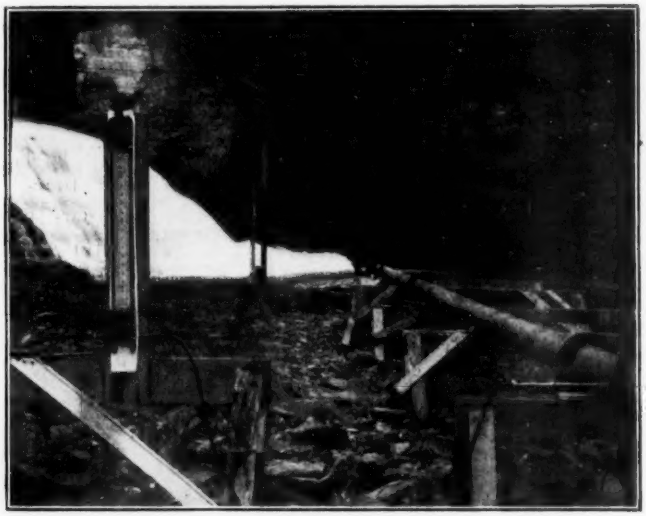


Fig. 3 Pitot Tube in Position for Horizontal Traverse in Straight Pipe and (at the Left) the Differential Gage

Fig. 4 Pitot Tube in Position for Vertical Traverse in Curve, Hose Connections and Differential Gage

11 In taking readings with the Pitot tube the instrument was carefully adjusted at the desired point on the diameter to be traversed, with the point opening facing the current. The tube was connected to one side of the differential water gage shown in Fig. 3 and Fig. 4, and the wall openings in the pipe were connected to the other side. All connections were made by means of small three-ply rubber hose. Great care was used to blow off all air from the hose connections. After the readings with the tube direct (i. e., pointing upstream), were taken, the tube was reversed to point downstream and another set of readings obtained.

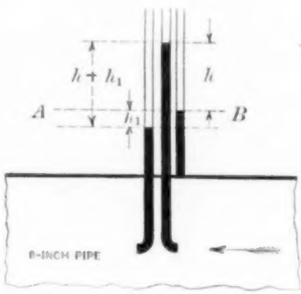


Fig. 5 Diagram of Pitot Tube Direct and Reverse

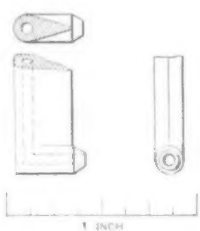


Fig. 6 Details of Pitot Tube

12 As a check on the Pitot tube measurements the quantity of water was measured by means of a brass nozzle at the end of the pipe line. The nozzle had been calibrated previously by tank measurement in the Cornell University Hydraulic Laboratory. To read the head on the nozzle a U tube mercury gage, open to the atmosphere on one side, was used.

13 By using the observed water gage difference when the point of the tube was facing the current as h, in the formula $v = c\sqrt{2gh}$, it was found that the value of c is unity, within a reasonable limit of error, as will be seen from the following table.

1	2	3	4	5	6
Date	Place of traverse	Mean velocity from point direct-wall reading. Pitot tube	Mean velocity from nozzle	Col. 3 / Col. 4	Ratio mean to center velocity. Pitot tube
Oct. 28, 1907	Vertical, 1.10 ft. upstream from curve.	16.66	16.73	0.996	0.8355
Oct. 29, 1907	Vertical, 1.10 ft. upstream from curve.	7.95	7.71	1.031	0.842
Oct. 31, 1907	Horizontal, 84.10 ft. downstream from curve	7.93	7.78	1.019	0.843
Nov. 4, 1907	Horizontal 84.10 ft. downstream from curve	8.39	8.24	1.018	0.850
Nov. 5, 1907	Horizontal, 38.0 ft. downstream from curve	7.83	7.92	0.989	0.832
Mean				1.011	0.8406

14 The results indicate an average error of about 1 per cent while the greatest error in any case is about 3 per cent. Since variations as wide as the above may be expected in work of this kind, even when conducted with the greatest care, the conclusion that the coefficient

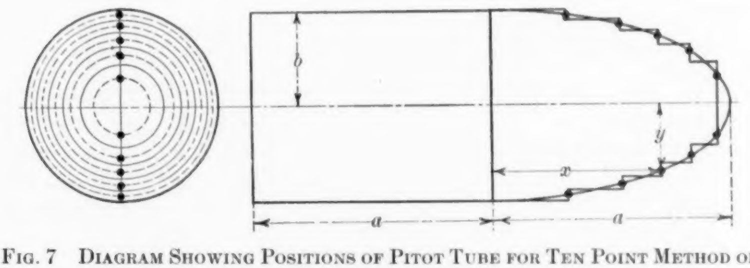


Fig. 7 Diagram Showing Positions of Pitot Tube for Ten Point Method of Traverse

of the tube is unity is justified. Probably a greater number of traverses would have given an average still nearer to unity.

The plottings of the traverses in the straight pipe are shown in Fig. 8.

15 Normal flow is shown by the shape of the curves and the ratios of mean to center velocities. (See preceding table.) The exact distance downstream from the curve where the distortion of velocities disappears is not known.[1] But it is shown by the traverse of November

[1] Saph and Schoder, Trans. Am. Soc. C. E., vol. 47, 1902, pp. 300–302.

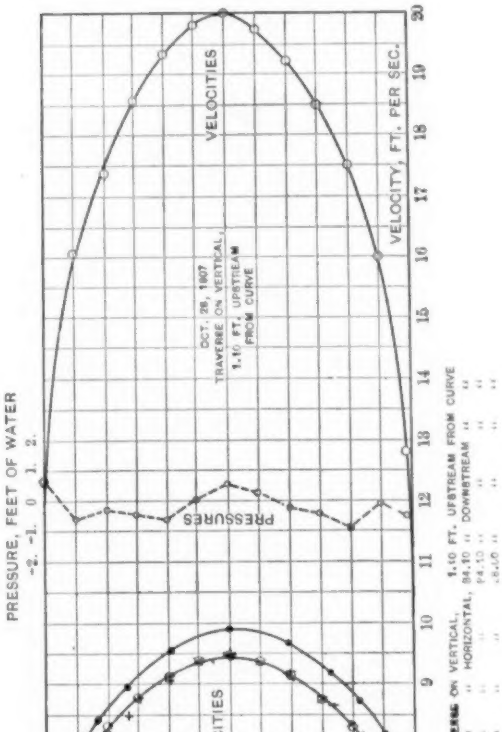


FIG. 8 DISTRIBUTION OF VELOCITIES AND PRESSURES IN A STRAIGHT PIPE

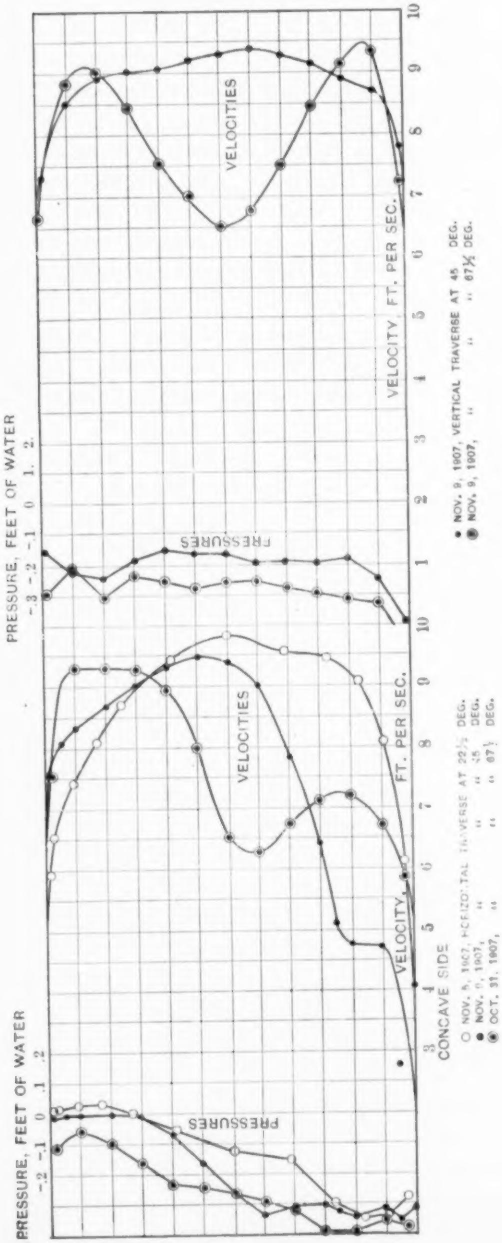


Fig. 9 Distribution of Velocities and Pressures in a Curved Pipe

5, 1907, that the flow is normal at a distance of 38 ft., or 76 diameters below the curve.

16 The gage readings obtained by reversing the tube have been investigated in connection with their relations both to the velocity and the pressure of the water at the various parts of the cross section of the pipe. The theory may be stated as follows:

17 At the right of Fig. 5 the Pitot tube is shown facing the current, while on the left the reversed position of the tube is shown. As already stated, a differential gage was used to read the quantities h and h_1, but for the sake of clearness, let open piezometers be imagined, so that the level of water as shown by the wall piezometer will be at the constant level A–B. As both h and h_1 are differences, it does not matter whether open piezometers or differential gages are used.

18 Let v be the true velocity for any point as obtained by the Pitot tube, point direct-wall reading.

Then $v = \sqrt{2\,g\,h}$

$v_1 = \sqrt{2\,g\,(h + h_1)}$, where h_1 = suction effect due to reversing the tube.

$v_1^2 = 2\,g\,h + 2\,g\,h_1 = v^2 + 2\,g\,h_1$

Therefore,

$$h_1 = \frac{v_1^2 - v^2}{2\,g}$$

But, from experiment,

$$v_1 = 1.133\,v$$

Therefore,

$$h_1 = \frac{(1.133^2 - 1)\,v^2}{2\,g} = \frac{(1.284 - 1)\,v^2}{2\,g} = 0.0044\,v^2$$

19 The value 1.133 is the mean obtained from the five straight pipe traverses. The individual results for each point of the traverses are given in Table 1. A summary follows:

Date	Average value of $c = \frac{v_1}{v}$
October 28, 1907	1.131
October 29, 1907	1.115
October 31, 1907	1.131
November 4, 1907	1.128
November 5, 1907	1.161
Mean	1.133

TABLE 1

Vertical Traverse 1.10 ft. Upstream from Curve. A Differential Mercury Gage was Used. The Differences in the Table are the Equivalent Water Differences

Entry on Top of Pipe — October 28, 1907

1 Distance in inches from bottom of pipe	2 Wall-point direct difference = h	3 Wall-point reversed, difference = h_1	4 $h + h_1$	5 $v = \sqrt{2gh}$	6 $v_1 = \sqrt{2g(h+h_1)}$	7 Ratio $\frac{v_1}{v}$	8 $0.0044 v^2$	9 Col. 8 − Col. 3
$\frac{1}{16}$	2.55	1.22	3.77	12.80	15.40	1.203	0.721	−0.50
$\frac{1}{2}$	3.97	1.27	4.24	16.00	16.52	1.032	1.127	−0.14
1	4.75	2.26	7.01	17.50	21.26	1.215	1.350	−0.91
$1\frac{1}{2}$	5.32	1.97	7.29	18.50	21.65	1.170	1.510	−0.46
2	5.75	1.90	7.65	19.22	22.20	1.155	1.630	−0.27
$2\frac{1}{2}$	6.04	1.49	7.53	19.72	22.02	1.117	1.710	+0.22
3	6.20	1.20	7.40	20.00	21.85	1.092	1.760	+0.56
$3\frac{1}{2}$	6.09	1.68	7.77	19.80	22.38	1.128	1.725	+0.04
4	5.80	2.33	8.13	19.32	22.90	1.185	1.640	−0.69
$4\frac{1}{2}$	5.34	2.00	7.34	18.55	21.75	1.172	1.520	−0.48
5	4.65	1.65	6.30	17.32	20.15	1.164	1.320	−0.33
$5\frac{1}{2}$	4.01	1.75	5.76	16.05	19.27	1.200	1.130	−0.62
6	2.36	−0.06	2.30	12.31	12.17	0.987	0.665	+0.60

Vertical Traverse 1.10 ft. Upstream from Curve. Water Differential Gage was Used

Entry on Top of Pipe — October 29, 1907

$\frac{1}{16}$	0.554	0.051	0.605	5.97	6.25	1.048	0.157	+0.106
$\frac{1}{2}$	0.902	0.223	1.125	7.62	8.50	1.115	0.255	+0.032
1	1.064	0.261	1.325	8.27	9.22	1.115	0.301	+0.040
$1\frac{1}{2}$	1.188	0.320	1.508	8.73	9.85	1.115	0.336	+0.016
2	1.295	0.360	1.655	9.12	10.32	1.133	0.366	+0.006
$2\frac{1}{2}$	1.365	0.350	1.715	9.36	10.50	1.120	0.386	+0.036
3	1.395	0.363	1.758	9.46	10.63	1.124	0.394	+0.031
$3\frac{1}{2}$	1.369	0.373	1.742	9.37	10.59	1.130	0.387	+0.014
4	1.290	0.338	1.628	9.10	10.23	1.126	0.364	+0.026
$4\frac{1}{2}$	1.198	0.347	1.545	8.77	9.97	1.137	0.338	−0.009
5	1.073	0.342	1.415	8.30	9.53	1.150	0.303	−0.039
$5\frac{1}{2}$	0.887	0.195	1.082	7.56	8.34	1.116	0.252	+0.057
6	0.515	0.072	0.587	5.76	6.15	1.069	0.146	+0.074

TABLE 1—Continued

HORIZONTAL TRAVERSE 1.05 FT. UPSTREAM FROM NOZZLE AND 84.11 FT. DOWNSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON LEFT HAND SIDE, LOOKING DOWNSTREAM OCTOBER 31, 1907

1	2	3	4	5	6	7	8	9
Distance in inches from bottom of pipe	Wall-point direct difference = h	Wall-point reversed, difference = h_1	$h + h_1$	$v = \sqrt{2gh}$	$v_1 = \sqrt{2g(h+h_1)}$	Ratio $\frac{v_1}{v}$	$0.0044\,v^2$	Col. 8 − Col. 3
$\frac{3}{32}$	0.588	0.238	0.776	6.15	7.07	1.150	0.166	−0.072
$\frac{1}{2}$	0.915	0.268	1.183	7.67	8.72	1.146	0.259	−0.009
1	1.080	0.308	1.388	8.32	9.45	1.135	0.305	−0.003
$1\frac{1}{2}$	1.195	0.342	1.537	8.76	9.94	1.133	0.338	−0.004
2	1.289	0.343	1.632	9.10	10.24	1.127	0.365	+0.022
$2\frac{1}{2}$	1.365	0.375	1.740	9.36	10.59	1.130	0.386	+0.011
3	1.392	0.380	1.772	9.45	10.68	1.130	0.393	+0.013
$3\frac{1}{2}$	1.362	0.372	1.734	9.36	10.54	1.127	0.386	+0.014
4	1.282	0.370	1.652	9.07	10.30	1.135	0.362	−0.008
$4\frac{1}{2}$	1.187	0.360	1.547	8.73	9.99	1.144	0.335	−0.025
5	1.043	0.343	1.386	8.18	9.43	1.152	0.295	−0.048
$5\frac{1}{2}$	0.880	0.198	1.078	7.52	8.31	1.105	0.248	+0.050
6	0.575	0.102	0.677	6.08	6.60	1.085	0.163	+0.061

HORIZONTAL TRAVERSE 1.05 FT. UPSTREAM FROM NOZZLE AND 84.11 FT. DOWNSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON LEFT HAND SIDE, LOOKING DOWNSTREAM NOVEMBER 4, 1907

TEN POINT METHOD. POINT NO. 1 IS OPPOSITE ENTRY

1	0.782	0.260	1.042	7.10	8.19	1.154	0.222	−0.038
2	1.041	0.295	1.336	8.19	9.27	1.133	0.295	0.000
3	1.180	0.305	1.485	8.72	9.77	1.118	0.334	+0.029
4	1.310	0.361	1.671	9.18	10.38	1.132	0.372	+0.011
5	1.452	0.408	1.860	9.67	10.94	1.133	0.411	+0.003
center	1.525	0.422	1.947	9.90	11.20	1.132	0.431	+0.009
6	1.417	0.375	1.792	9.54	10.74	1.129	0.400	+0.025
7	1.241	0.375	1.616	8.94	10.20	1.142	0.352	−0.023
8	1.095	0.380	1.475	8.40	9.75	1.160	0.311	−0.069
9	0.930	0.190	1.120	7.74	8.50	1.100	0.263	+0.073
10	0.671	0.105	0.776	6.58	7.07	1.074	0.191	+0.086

TABLE 1—Continued

HORIZONTAL TRAVERSE 38.0 FT. DOWNSTREAM FROM CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON LEFT HAND SIDE, LOOKING DOWNSTREAM — NOVEMBER 5, 1907

TEN POINT METHOD. POINT NO. 1 IS OPPOSITE ENTRY

1	2	3	4	5	6	7	8	9
Distance in inches rom bottom of pipe	Wall-point direct difference = h	Wall-point reversed, difference = h_1	$h + h_1$	$v = \sqrt{2gh}$	$v_1 = \sqrt{2g(h+h_1)}$	Ratio $\frac{v_1}{v}$	$0.0044\,v^2$	Col. 8 − Col. 3
1	0.558	0.230	0.788	6.00	7.12	1.186	0.158	−0.072
2	0.902	0.314	1.216	7.62	8.85	1.162	0.255	−0.059
3	1.033	0.344	1.377	8.15	9.42	1.155	0.292	−0.052
4	1.170	0.382	1.552	8.68	10.00	1.150	0.332	−0.050
5	1.307	0.439	1.746	9.17	10.62	1.156	0.370	−0.069
center	1.388	0.459	1.847	9.45	10.90	1.154	0.393	−0.066
6	1.262	0.433	1.695	9.02	10.45	1.158	0.358	−0.075
7	1.115	0.404	1.519	8.47	9.88	1.167	0.316	−0.088
8	1.000	0.422	1.422	8.02	9.57	1.194	0.283	−0.039
9	0.851	0.257	1.108	7.41	8.45	1.139	0.242	−0.015
10	0.557	0.190	0.747	5.99	6.93	1.155	0.180	−0.010

20 It will be seen from the calculated values for $\frac{v_1}{v}$ in Table 1, that for any particular traverse the ratio is practically constant across the pipe, except near the wall where eddies and disturbances tend to vitiate observations.

21 If now we calculate backwards and subtract from h_1 the quantity $0.0044v^2$, we will get zero on the average. The variation from zero for the five straight pipe traverses is shown in Table 1 and in Fig. 8 at the left.

22 This average value of zero across the pipe diameter means that the pressure throughout the cross section is identical with that at the wall (the effect of hydrostatic differences of level, of course, excepted).

23 The quality of pressure throughout the cross section of a straight pipe in which water is flowing may be inferred from the fact that a Pitot tube having a point opening only, and taking its reference pressure at the wall of the pipe, gives the true mean velocity according to the formula,

$$v = \sqrt{2gh}$$

TABLE 2

HORIZONTAL TRAVERSE, IN CURVE AT 45 DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON CONCAVE SIDE — NOVEMBER 9, 1907

1	2	3	4	5	6	7	8
Distance in inches from side opposite entry	Wall-point direct difference $= h$	Wall-point reverse, difference $= h_1$	$h + h_1$	$v = \frac{v_1}{1.133} = 0.883\, v_1$ $= 0.883 \sqrt{2g(h+h_1)}$	$v_1 = \sqrt{2g(h+h_1)}$	$0.0044\, v^2$	Col. 7 − Col. 3
1/16	0.870	0.271	1.141	7.58	The ratio $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.253	−0.018
1/4	1.005	0.293	1.298	8.08		0.288	−0.005
1/2	1.080	0.300	1.380	8.33		0.306	−0.006
1	1.170	0.335	1.505	8.68		0.333	−0.002
1 1/2	1.250	0.372	1.622	9.03		0.360	−0.012
2	1.277	0.453	1.730	9.31		0.382	−0.071
2 1/2	1.223	0.563	1.786	9.48		0.396	−0.167
3	1.103	0.653	1.756	9.39		0.389	−0.264
3 1/2	0.920	0.695	1.615	9.00		0.357	−0.338
4	0.644	0.583	1.227	7.85		0.272	−0.311
4 1/2	0.340	0.485	0.825	6.44		0.183	−0.302
4 3/4	0.083	0.437	0.520	5.11		0.115	−0.322
5	0.005	0.447	0.452	4.76		0.100	−0.347
5 1/2	0.030	0.413	0.443	4.72		0.098	−0.315
5 3/4	−0.245	0.390	0.155	2.79		0.034	−0.356
6	−0.057	0.387	0.330	4.07		0.073	−0.314

HORIZONTAL TRAVERSE, IN CURVE AT 67½ DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON CONCAVE SIDE — OCTOBER 31, 1907

1	2	3	4	5	6	7	8
1/8	0.780	0.360	1.140	7.55	The ratio $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.248	−0.112
1/2	1.280	0.441	1.721	9.30		0.381	−0.060
1	1.250	0.477	1.727	9.31		0.381	−0.096
1 1/2	1.167	0.545	1.712	9.28		0.379	−0.166
2	1.012	0.583	1.593	8.94		0.352	−0.231
2 1/2	0.743	0.525	1.278	8.01		0.282	−0.243
3	0.395	0.450	0.845	6.52		0.187	−0.263
3 1/2	0.323	0.465	0.788	6.29		0.174	−0.291
4	0.373	0.523	0.896	6.71		0.198	−0.325
4 1/2	0.400	0.613	1.013	7.13		0.223	−0.390
5	0.415	0.622	1.035	7.21		0.228	−0.394
5 1/2	0.340	0.555	0.895	6.71		0.198	−0.357
5 13/16	0.160	0.527	0.687	5.87		0.152	−0.375

TABLE 2—Continued

VERTICAL TRAVERSE IN CURVE, 45 DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY AT TOP OF PIPE — NOVEMBER 9, 1907

1	2	3	4	5	6	7	8
Distance in inches from side opposite entry	Wall-point direct difference $= h$	Wall-point reverse, difference $= h_1$	$h + h_1$	$v = \frac{v_1}{1.133} = 0.883\, v_1 = 0.883\sqrt{2g\,(h + h_1)}$	$v_1 = \sqrt{2g\,(h + h_1)}$	$0.0044\, v^2$	Col. 7 − Col. 3
1/16	0.556	0.655	1.211	7.80	The ratio $\frac{v_1}{v}$ as obtained in the straight pipe line was used to compute v in the curve.	0.268	−0.387
½	0.936	0.580	1.516	8.71		0.334	−0.246
1	1.041	0.534	1.575	8.90		0.349	−0.185
1½	1.109	0.563	1.672	9.15		0.369	−0.194
2	1.147	0.568	1.715	9.28		0.379	−0.189
2½	1.163	0.580	1.743	9.35		0.385	−0.195
3	1.174	0.542	1.716	9.28		0.379	−0.163
3½	1.151	0.535	1.686	9.19		0.372	−0.163
4	1.120	0.513	1.633	9.05		0.360	−0.153
4½	1.077	0.540	1.617	9.01		0.357	−0.183
5	0.990	0.590	1.580	8.90		0.349	−0.241
5½	0.897	0.543	1.440	8.50		0.318	−0.225
5⅞	0.677	0.390	1.067	7.31		0.235	−0.155

VERTICAL TRAVERSE IN CURVE, 67½ DEG. FROM UPSTREAM END OF CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY AT TOP OF PIPE — NOVEMBER 9, 1907

1/16	0.335	0.707	1.042	7.23	$v = \frac{v_1}{1.133} = 0.883\, v_1$ The ratio of $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.230	−0.477
½	1.027	0.713	1.740	9.34		0.384	−0.329
1	0.973	0.680	1.653	9.11		0.365	−0.315
1½	0.820	0.607	1.427	8.45		0.315	−0.292
2	0.598	0.523	1.121	7.50		0.248	−0.275
2½	0.460	0.453	0.913	6.77		0.202	−0.251
3	0.402	0.448	0.850	6.53		0.188	−0.260
3½	0.487	0.493	0.980	7.01		0.216	−0.277
4	0.630	0.503	1.133	7.54		0.251	−0.252
4½	0.870	0.550	1.420	8.44		0.314	−0.236
5	0.960	0.660	1.620	9.01		0.357	−0.303
5½	1.008	0.543	1.551	8.82		0.342	−0.201
6	1.400	0.486	0.886	6.66		0.195	−0.291

TABLE 2—Continued

HORIZONTAL TRAVERSE IN CURVE, 22½ DEG. FROM UPSTREAM END OF THE CURVE. WATER DIFFERENTIAL GAGE WAS USED

ENTRY ON CONCAVE SIDE NOVEMBER 5, 1907

TEN POINT METHOD. POINT NO. 1 IS OPPOSITE ENTRY

1	2	3	4	5	6	7	8
Distance in inches from side opposite entry	Wall-point direct difference = h	Wall-point reverse, difference = h_1	$h + h_1$.	$v = \frac{v_1}{1.133} = 0.883\, v_1 = 0.883\sqrt{2g(h+h_1)}$	$v_1 = \sqrt{2g(h+h_1)}$	$0.0044\, v^2$	Col. 7 − Col. 3
1	0.685	0.165	0.850	6.54	$v = \frac{v_1}{1.133} = 0.883\, V_1$ The ratio $\frac{v_1}{v}$ as obtained in the straight pipe was used to compute v in the curve.	0.188	+013.0
2	0.880	0.215	1.095	7.42		0.242	+0.027
3	1.055	0.250	1.305	8.10		0.282	+0.032
4	1.170	0.332	1.502	8.70		0.333	+0.001
5	1.315	0.447	1.762	9.41		0.390	−0.057
center	1.378	0.548	1.926	9.83		0.426	−0.122
6	1.272	0.558	1.830	9.59		0.405	−0.153
7	1.090	0.695	1.785	9.47		0.395	−0.300
8	0.935	0.710	1.645	9.09		0.364	−0.346
9	0.687	0.630	1.317	8.10		0.289	−0.341
10	0.307	0.445	0.752	6.14		0.166	−0.279

This has been shown for many different sizes of pipes and for a wide range of velocities by several experimenters. If the pressure varied across the pipe this relation would not hold. Further direct experimental proof has been given by at least two observers.[1]

24 Having calculated the value $h + h_1$, from the observations on the curve traverses, we may obtain the true velocity at any point by using the relation

$$\frac{v_1}{v} = \frac{\sqrt{2g\,(h+h_1)}}{\sqrt{2\,g\,h}} = 1.133$$

and having the true velocity, we calculate the corresponding suction head by using the formula

$$h_1 = 0.0044\, v^2$$

If now for each point on the traverse this calculated suction head be subtracted algebraically from the corresponding gage difference with

[1] Note by M. Bazin, Trans. Am. Soc. C. E., vol. 47, 1902, p. 248, and The Pitot Tube, W. B. Gregory, Trans. Am. Soc. M. E., vol. 25, p. 201.

the tube reversed, we have an indication of the pressure variation across the pipe diameter. The results of the curve traverses are plotted in Fig. 9.

25 The horizontal traverses in the curve show the gradual swinging of the maximum velocity toward the convex side of the curve as the water passes around, except in the case of the traverse at 22½ deg. which shows a tendency in the opposite direction. The vertical traverses show nearly symmetrical distribution above and below the center.

26 The plottings of the pressure differences, as explained above, show that the pressure is highest near the convex side of the pipe and least near the concave side. The actual quantitative results are complicated by the loss of head between the upstream piezometer and the points in the curve where the traverses were made. But this is a constant and the qualitative results are not altered. The high pressure at the convex side of the curve may be explained by the centrifugal action of the water.

27 In the case of the pressures for the vertical traverses, no corresponding change of pressure with velocity is noticeable.

28 In using the Pitot tube to obtain the mean velocity of water flowing in a straight pipe a question arises as to the number of points of observation necessary that the resulting error may be reasonably small.

29 The computations given below are based on the theory that the velocities in the various parts of a cross section of a straight pipe in which water is flowing are arranged in such a way that they form an ellipsoid of revolution, with the velocity at the center equal to twice the velocity at the walls of the pipe. On any diameter the plotting of velocities parallel to the axis of a straight pipe may be thought of as made up of a semi-ellipse joined to a rectangle of length equal to the semi-ellipse.[1]

30 Suppose the cross section of the pipe to be divided by concentric circles having areas, respectively, of 1/20, 3/20, 5/20, 7/20, 9/20, 11/20, 13/20, 15/20, 17/20 and 19/20 of the cross sectional area. If in making a traverse on a diameter, observations be taken on the circumferences of these circles, there will be 20 readings for a traverse, and the mean of the velocities found from these 20 readings will be the mean velocity of the fluid. In order to investigate this theory, let us assume an ellipse having one-half of its major and minor axes

[1] The accompanying curves confirm this assumption. For similar work, see bibliography in Engineering News, December 21, 1905.

denoted by a and b, respectively. Let x and y be the coördinates of any point on the ellipse. The results are arranged in tabular form in Table 3.

TABLE 3

No	Area	y	x
1	1/20	0.2236b	0.9747a
2	3/20	0.3873b	0.9220a
3	5/20	0.5000b	0.8660a
4	7/20	0.5916b	0.8062a
5	9/20	0.6708b	0.7416a
6	11/20	0.7416b	0.6708a
7	13/20	0.8062b	0.5916a
8	15/20	0.8660b	0.5000a
9	17/20	0.9220b	0.3873a
10	19/20	0.9747b	0.2236a
		Mean =	0.6684

Volume of true ellipsoid with cylinder $= 1.6667\pi\, a\, b^2$
Volume by above method $= 1.6684\pi\, a\, b^2$

THE EQUATION OF THE ELLIPSE IS

$$\frac{x^2}{a^2} + \frac{y^2}{b^2} = 1$$

TABLE 4

No.	Area	y	x
1	1/10	0.3162b	0.9487a
2	3/10	0.5477b	0.8367a
3	5/10	0.7071b	0.7071a
4	7/10	0.8367b	0.5477a
5	9/10	0.9487b	0.3162a
		Mean =	0.6713a

$$\frac{1.6713}{1.6667} = 1.0028$$

31 It is seen that there is practically no error in calculating the mean velocity from a traverse with 20 points as indicated above. It will be noted that the center velocity is omitted in the calculation.

32 Next let us investigate the error when only ten points are used, as shown in Fig. 7. In this case the areas of the concentric circles will be 1/10, 3/10, 5/10, 7/10, and 9/10. The results are given in Table 4. The error in this case is only 0.28 of 1 per cent, which is well within the error of observation. In this case also the center velocity is not used.

33 The Pitot tube used by the writers in this investigation, is not of the form that they would recommend for general use in cases of distorted flow, although it is well adapted to traversing straight pipes where a wall piezometer gives correct mean pressure indications. It was the only available tube suitable for the pipe. In general where a wall piezometer is not available, a tube with both impact and static openings would be more convenient, and less liable to errors of observation, and would effect a material saving of time.

34 The method above described, of obtaining the true velocity at any point by means of the tube used, corresponds to working with a

tube having both openings. In the latter case it is, of course, unnecessary to reverse the tube. The corresponding suction, if there be any, on the static openings may be obtained by using a wall piezometer in rating the tube.[1] Having rated the tube, either with the wall piezometer or by tank measurement of the discharge, the wall piezometer is no longer necessary to give pressure indications. A tube may be so designed that there will be no suction action at the static openings, i. e., the static openings will act precisely like wall piezometers and will give the true pressure. In this case the coefficient of the tube is unity.

35 In case the coefficient of the tube is not unity, and the impact point is of proper shape, the following statement gives the method of obtaining the suction head at the static openings.

36 In general we will have for the formula to be used with any Pitot tube,

$$v = c\sqrt{2\,g\,H}$$

where

$$H = (h + h_1)$$

That is, H is the water column difference between the impact and static openings.[2] Then

$$v^2 = c^2\,2\,g\,H, \text{ or } H = \frac{v^2}{c^2\,2\,g}$$

37 The impact opening of the tube gives the head

$$h = \frac{v^2}{2\,g}$$

The static openings give the head

$$h_1 = (H - h) = \left(\frac{1}{c^2} - 1\right)\frac{v^2}{2\,g}$$

38 Thus, if the rating of a two-opening tube shows a coefficient of 0.883, we have

$$h_1 = \left(\frac{1}{0.883^2} - 1\right)\frac{v^2}{2\,g} = (1.284 - 1)\frac{v^2}{2\,g} = 0\ 284\,\frac{v^2}{2\,g}$$

[1] Water Measurements in Connection with a Test of a Centrifugal Pump at Jourdan Avenue Drainage Station, New Orleans, La

[2] See article by W. M. White, Journal of Association of Engineering Societies, August 1901, p. 64

CONCLUSION

39 To sum up:

a A method has been given for determining, by means of a Pitot tube with a single opening, the variation of velocities across a curved passage in which water is flowing.

b The variation of pressures across a curved passage has been determined by means of a Pitot tube with a single opening. So far as the writers know, this method has never been suggested before.

c A method has been pointed out by which, with a tube having both impact and static openings, the pressure at any point across a curved passage may be determined.

d By inference, the Pitot tube offers a solution of problems in the perfecting of the designs of turbines, pumps, etc., where it is important to know precisely how the fluid flows through the curved passages, especially how the velocities and pressures vary.

DISCUSSION

Mr. G. A. Orrok I have been greatly interested in the work done with Pitot tubes by Professors Schoder and Gregory, Mr. White, and the other gentlemen who have been experimenting with them. The curves given by the authors of the paper show in a very good way the eddies and other peculiarities in the flow of the water around bends. In making some power plant tests I have had occasion to measure the condensing water by the use of Pitot tubes placed in the condenser discharge pipe. This pipe was 36 in. in diameter with a number of bends and no piece of straight pipe of sufficient length to free the current of water from eddies. In making the traverses I obtained curves which were almost identical with the curves shown by the authors.

2 Referring to the use of upstream and downstream Pitot tubes I have attempted to use them, but with negative results, it being almost impossible to secure anything that was consistent; this probably because of the difficulty of keeping the two tubes directly opposite and in line with the flow of the water. I got much better results by using a Pitot tube with a long nozzle in the line of flow, the velocity opening being a surface of revolution, and the static opening four longitudinal slots, 90 deg. apart in the surface of the outer pipe.

3 The constant of this tube I find to be almost invariably, within limits, 100 per cent.

Dr. Sanford A. Moss I have been greatly interested in the portion of this paper referring to use of "downstream readings" of a single opening pitot tube. If it is possible to formulate a definite law between "impact head" (difference between "upstream reading" and "static pressure") and "downstream head" (difference between "static pressure" and pressure shown by tube pointing down stream), the method of using a direct and reversed tube for computation of actual velocity and actual static pressure would be very useful in hydrodynamic work of all kinds. I believe a great deal of experimental work must be done before a definite law can be formulated in the matter. I believe that the relation between "impact head" and "downstream head" may vary with the shape of the impact tube, density and velocity.

2 Some time ago I planned to use upstream and downstream readings for measurement of flow of air in straight pipes for cases where there were irregularities due to previous elbows, etc., the idea being to obtain true velocity and static pressure by exactly the method used in the paper.

3 Experiments to determine the coefficient were made by use of an impact tube pointing first upstream and then downstream in a jet escaping from an open pipe into the atmosphere. In such a case if the observations are taken at a little distance from the end of the open pipe, the static pressure is known to be atmospheric.

4 The relation between the upstream head and the downstream head varied with the velocity to an extent which was not precisely determined. Most important, however, was the fact that *the downstream head reading was greatly affected by the shape of the tube.* As is well known, the upstream reading, or true impact reading, always gives the velocity head regardless of shape of tube. This certainly does not apply to the downstream reading. A similar experiment was made with a small impact tube in a steam jet of high velocity. The downstream head in this case was much smaller than in other cases.

5 It also seems probable that the downstream reading is greatly affected by slight angularity of the stream lines caused by local eddies, etc. It is certainly true that the static opening or side opening of an ordinary pitot tube is similarly affected. This seems to indicate that a downstream tube, or a side opening for measurement

of static pressure, is unsatisfactory, except for parallel flow in a straight pipe. However, in such a case, it seems tolerably certain that the static pressure throughout is the same as the pressure measured by a hole in the pipe wall.

6 The Pitometer Company have developed a system for measurement of flow of water in pipes from sum of downstream and upstream heads. They use an experimentally determined coefficient for multiplying velocity as computed from total head, to obtain the true velocity. This coefficient was originally 0.8, but I am informed by Mr. Cole that 0.84 is shown by recent experiments to be a more accurate value. The reciprocal of this, 1.19, corresponds to the value C in the present paper given as 1.133. There is here a discrepancy of 6 per cent. The values of this ratio C, given in the present paper, seem quite discordant, and values from 1.09 to 1.21 occur in the region away from the edges of the pipe.

7 In my own work in this matter, instead of using C, the ratio of velocity computed from sum of upstream and downstream heads, to the actual velocity, I have used the direct ratio of downstream head to upstream or true velocity head. This is $C^2 - 1$. This is the ratio between the directly observed quantities, and different values of it indicate the direct uncertainty in the matter, better than the different values of C which have a much smaller percentage difference. Various values of this direct head ratio are given in the accompanying table.

TABLE OF RATIOS OF DOWN STREAM HEAD TO IMPACT HEAD

(static head − downstream head) ÷ (Impact or upstream head − static head)	
Value corresponding to $C = 1.133$ of present paper	0.28
Value corresponding to Pitometer coefficient $1/C = 0.84$	0.42
Value corresponding to Pitometer coefficient $1/C = 0.80$	0.56
Value obtained by writer with air	0.25 to 0.40
Value obtained by writer with high velocity stream	0.28

Prof. H. T. Eddy[1] I fail to see how it is theoretically possible that the readings of a Pitot tube, directed downstream, should have any necessary relationship to the velocity of the stream; or be of any importance. It is theoretically and experimentally known that the readings of a Pitot tube directed upstream will increase with every increase of stream velocity, however great. But when directed downstream there is a certain velocity of stream which would in any given case produce a suction head as great as the depth of immersion,

[1] Member of the Gas Power Section.

and any stream velocity greater than that would draw air into the water through the tube, thus making it impossible to read any of these greater velocities by a tube directed downstream.

2 Somewhat the same phenomenon is observed in the wake of a ship, where there is a depression of the water at the stern. No matter what the speed of the ship, the pressure at the stern cannot be less than that of the atmosphere. It would seem to me then, that downstream readings of the Pitot tube must, in general, from the nature of the case, be illusory. There is a possibility however that for any low velocities a Pitot tube might be so rated experimentally as to make such readings of some value.

THE AUTHORS There is abundant experimental evidence showing that the suction effect due to reversing a Pitot tube is considerably affected by the shape of the tube, as Dr. Moss states in his discussion.

2 However, the writers would hardly designate the downstream readings as "illusory," to use Professor Eddy's adjective. No doubt there is mystery in all of nature's phenomena. Moreover, in the experiments described in the paper there are noticeable individual departures (in column 7 of Table 1) from the average value 1.133 for $\frac{v_1}{v_1}$. Still the mean value for any one of the five traverses is within less than 3 per cent of 1.133. This speaks for itself.

3 Professor Eddy's statement concerning depth of immersion and the drawing of air into the downstream opening of the Pitot tube may have some application to open channel conditions near the surface, but is not applicable to a Pitot tube in a pipe under pressure. There seems to be no reason why the suction head should not vary as a function of the velocity far beyond the range investigated by the writers. This includes velocities as high as 20 ft. per second. The limiting value in any extreme case would occur when a vacuum would be produced at the downstream opening. But for all velocities met with in practice the experimental evidence indicates that in closed pipes the suction head does vary as a definite function of the velocity.

No. 1200

COMPARISON OF SCREW THREAD STANDARDS

By Amasa Trowbridge, Hartford, Conn.

Member of the Society

The accompanying diagram is submitted to show what is possible or impossible in combining the present recognized standards for screw threads. The United States standard thread has long been considered too coarse for diameters of $\frac{3}{8}$ in. and smaller. The dash and dot curve of the diagram is for the Pratt & Whitney standard thread for machine screws and is substantially what most shops use for screws having diameters in fractions. As this standard nearly coincides with the standard recommended by the Machine Screw Committee of The American Society of Mechanical Engineers, which gives better and more uniform proportions, it should be discarded wherever possible in favor of the latter. Likewise, it seems advisable to stop the United States standard at a diameter of $\frac{7}{16}$ in. instead of at a diameter of $\frac{1}{4}$ in., following the new standard of the Committee of the A. S. M. E., for sizes below $\frac{7}{16}$ in. diameter and inserting suitable extra sizes in this new standard to care for such fractional sizes as are not at present included in it.

2 It is apparent from the figure that the United States standard is better from $\frac{7}{16}$ in. up than the A. S. M. E. sizes, hence there is nothing to be gained by continuing the latter above this point.

3 To manufacturers of parts or accessories for automobiles, the problem of using the standard of the Association of Licensed Automobile Manufacturers presents itself. Inspection of the diagram shows that this standard is not adapted for combining with the other standards. Also, while it is undoubtedly very good for use where excessive vibration is encountered, it is not suitable for general use. The Veeder Manufacturing Co., Hartford, Conn., furnishes nearly all the odometers used on automobiles and at the same time manufactures large numbers of counters for use on machinery, and they have had to consider all these standards. This diagram was used to study the

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

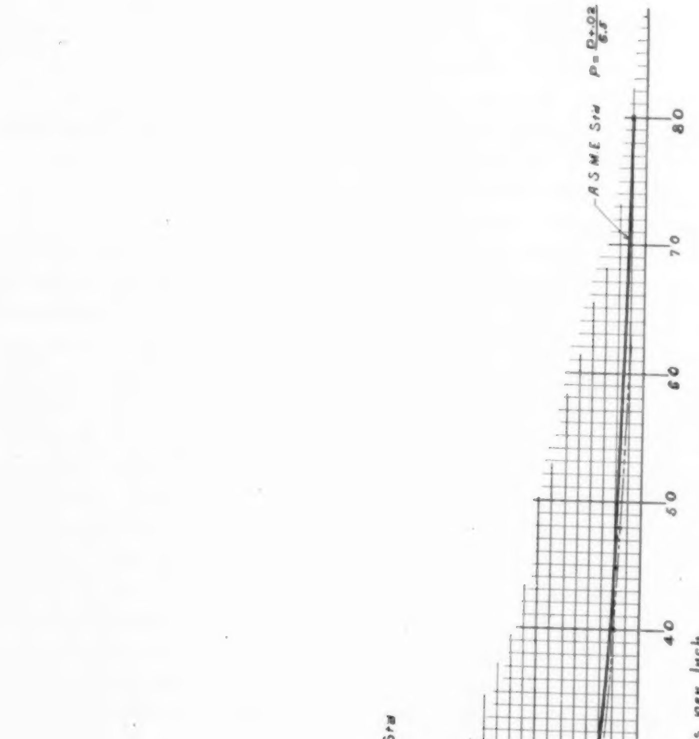


FIG. 1 COMPARISON OF SCREW THREAD STANDARDS

problem and as it may be of service to others, it is brought to the attention of the Society.

DISCUSSION

MR. LUTHER D. BURLINGAME There seems to be a persistent demand for screw threads of a finer pitch than those derived from the formula for the U. S. or Franklin Institute standard.

2 This demand has been met by the automobile manufacturers by the adoption of a standard having radically finer pitches than the U. S. standard. This new standard, however, does not meet the more important needs of machine tool builders and other manufacturers having similar conditions to deal with, for use as a standard for bolts, screws, studs, etc., as the thread is too fine where used in castings as well as in steel.

3 That the U. S. standard has too coarse a thread to give the best results on the class of work just referred to has been repeatedly pointed out. The Brown & Sharpe Manufacturing Company discarded it 30 years ago and have since used a standard of their own with a finer pitch covering sizes from 1 in. diameter down to $\frac{3}{32}$ in., the pitch of the threads being intermediate between the U. S. standard and the new automobile standard, called the A.L.A.M. standard.

4 In the discussion of the paper by Mr. Chas. C. Tyler at the Boston meeting of the Society in 1902, at least five different formulae were suggested for obtaining finer threads. All those taking part in the discussion favored finer threads for the small sizes of screws. Mr. Wilfred Lewis said at that time: "The pitch of the standard $\frac{1}{4}$ in. screw is generally admitted to be too coarse and many taps and dies for this size are now made with 24 threads to the inch instead of 20." The pitches recommended by the Committee on Standard Proportions for Machine Screws are a good average of the pitches up to and including $\frac{5}{16}$ in. diameter recommended by the different authorities at the Boston meeting. That these pitches of the A. S. M. E. standard come midway between those in use by the Brown & Sharpe and Pratt & Whitney companies for the small sizes from $\frac{1}{4}$ in. down to $\frac{3}{32}$ in. is a good indication that they give a fair average.

5 The suggestion now made in Mr. Trowbridge's paper that the A. S. M. E. standard for pitches should be generally adopted for these small screws seems in the right direction. He also points out the objection to the U. S. standard as being too coarse for sizes below $\frac{7}{16}$ in. and recommends a standard that is finer.

6 Mr. Chas. T. Porter has consistently and ably urged the adoption of finer threads and gave his reasons in a paper presented before the Society at the New York meeting in 1902 and has continued to urge it since. Arguments in favor of using a finer thread for the regular machine screws up to 1 in. are that the screw is stronger and the tapping cheaper.

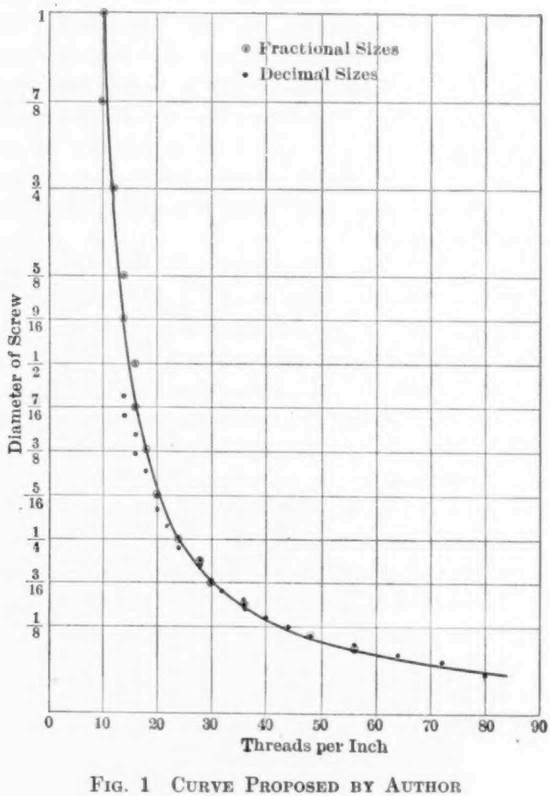


FIG. 1 CURVE PROPOSED BY AUTHOR

7 The British Engineering Standards Committee have, within a few years, adopted a standard for fine threads to provide for the same need that has been pointed out as existing in this country.

8 With Mr. Trowbridge's suggestion as a starting point, it seemed possible that a formula might be devised that would follow the A. S. M. E. standard as far as the $\frac{5}{16}$ in. size and for larger sizes give a new series of pitches intermediate between the U. S. standard and the

A.L.A.M. standard, thus, by making one formula cover the whole ground, "killing two birds with one stone," as it were.

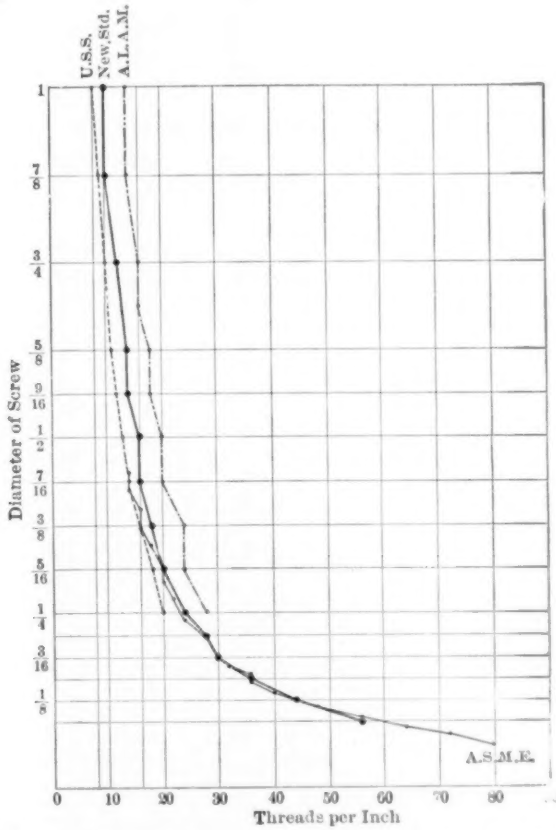


Fig. 2 Curve for Pitches Intermediate Between U. S. and A. L. A. M Standards

9 I have succeeded in deriving such a formula and it produces the curve shown in diagram No. 1, the formula being

$$N = 5.5 + \frac{4.7}{D} \text{ or } P = \frac{D}{5.5\,D + 4.7}$$

When N = the number of threads per inch
and D = the outside diameter of the screw.

10 This curve follows the pitches of the A. S. M. E. standard up to $\frac{5}{16}$ in. diameter as closely as the curve produced by their published formula. That this is possible is due to the fact that the committee departed from their exact formula sufficiently to give even pitches and still further to retain some of the pitches already in use.

TABLE 1 COMPARATIVE TABLE OF PITCHES

DIAM.	NO. OF THREADS PER INCH							
	U. S. Std.	Whitworth std.	British Fine std.	A. L. A. M. std.	B. & S. std.	Bond std.	Lewis std.	Proposed intermediate standard
$2\frac{3}{4}$	4	$3\frac{1}{2}$	6					
$2\frac{1}{2}$	4	4	6					
2	$4\frac{1}{2}$	$4\frac{1}{2}$	7					
$1\frac{1}{2}$	6	6	8					
$1\frac{1}{4}$	7	7	9					
1	8	8	10	14	10	8.5	7.3	10
$\frac{7}{8}$	9	9	11	14	10			10
$\frac{3}{4}$	10	10	12	16	12			12
$\frac{5}{8}$	11	11	14	18	12			14
$\frac{9}{16}$	12	12	16	18	14			14
$\frac{1}{2}$	13	12	16	20	14	14.8	12	16
$\frac{7}{16}$	14	14	18	20	14			16
$\frac{3}{8}$	16	16	20	24	16			18
$\frac{5}{16}$	18	18	22	24	20			20
$\frac{1}{4}$	20	20	25	28	24	25	22	24
$\frac{7}{32}$								28
$\frac{3}{16}$								30
$\frac{5}{32}$								36
$\frac{1}{8}$								44
$\frac{3}{32}$								56

11 Commencing at $\frac{1}{4}$ in. the new formula gives a series of pitches intermediate between the U. S. standard and the A. L. A. M., as shown in diagram No. 2 and the accompanying table. If it should not be considered objectionable to use odd numbers of threads, the series would follow the formula almost exactly. My preference, however, would be to depart from the formula to the extent necessary to keep the pitches in even figures, as was done when the A. S. M. E. and A. L. A. M. standards were established.

FORMULAE

U. S. Standard

$$P = \frac{\sqrt{16\,D + 10} - 2.909}{16.64} \text{ or}$$

$$P = 0.24\,\sqrt{D + 0.625} - 0.175$$

British fine standard

Up to 1 inch in diameter

$$P = \frac{\sqrt[3]{D^2}}{10}$$

Above 1 inch in diameter

$$P = \frac{\sqrt[8]{D^5}}{10}$$

Bond standard

$$P = 0.23\,\sqrt{D + 0.625} - 0.175$$

Lewis standa d

$$P = 0.22\,\sqrt{D + 0.25} - 0.11$$

Proposed intermediate standard

$$N = 5.5 + \frac{4.7}{D} \text{ or}$$

$$P = \frac{D}{5.5\,D + 4.7}$$

12 The new formula here presented gradually diverges from the A. S. M. E. standard for sizes larger than $\frac{5}{16}$ in., for that standard soon crosses the line of the U. S. standard and becomes even coarser than the U. S. standard beyond $\frac{7}{16}$ in. as seen in Mr. Trowbridge's diagram.

13 The proposed standard has this to commend it, that it is simply an extension of an already established standard rather than the adding of an entirely new one. It is one that would give a proper proportion of diameter to pitch for sizes from the smallest watch screws up to the largest bolts and screws.

14 While on general principles I hesitate to suggest anything which would seem to add to the number of standards for screws, I feel that sooner or later this problem will have to be dealt with and if it is authoritatively done by the Society, it will be settled once for all.

Mr. F. A. Halsey There is no doubt but that there is need of a standard for fine screw threads. Under the present conditions, as time goes on, people will adopt standards of their own, and eventually

there will be great diversity. Under these circumstances it seems wise to take action on this matter and establish a standard that will be generally adopted. Mr. Burlingame has opened the subject, and I move that a committee be appointed to consider the matter. [The motion was seconded and carried.—EDITOR.]

MR. CHAS. T. PORTER Three-quarters of a century ago in England machine screw threads were in a state of chaos. Each builder of machinery had a system of his own. This condition was cherished by these builders because it rendered it necessary that the customers of each one should come to him for their repairs.

2 The man for the hour was Joseph Whitworth. His comprehensive mind formed the really grand conception of a uniform or universal system of machine screw threads, a feature the value of which is incalculable. In order to procure the adoption of his system, he found it necessary to make it the mean of all existing systems. I was told by Mr. Whitworth himself that but for this feature he would never have succeeded in procuring its adoption. So the Whitworth thread was a compromise, but he established the one thing of supreme importance, which was, uniformity.

3 The Whitworth thread was copied by American mechanics without change so far as the pitches, or numbers of threads to the inch, were concerned. In the form of the thread, the Americans made two changes, one a step forward, the other a long step backward. The first was, changing the angle of the thread from 54 deg. to 60 deg., an angle which can be readily originated and verified with accuracy, and can also be bisected, assuring the same angle to each side of the thread, both which points were extremely difficult with the 54-deg. angle. The step backward was, abandoning the round top and bottom of the Whitworth thread, and continuing the sides of the thread to a sharp edge at the top and bottom.

4 In the year 1864, Mr. William Sellers presented to the Franklin Institute in Philadelphia his important and now familiar improvement, of making the top and bottom of the thread flat surfaces, the width of which is one-eighth of the pitch.

5 In 1868, on returning from England and commencing the manufacture of the Allen Engine in a little shop in Harlem, I adopted the Sellers thread, obtaining the necessary hobs from the firm of Wm. Sellers & Co. I also imported from England a set of cylindrical gauges, and a remarkable bolt-threading machine, both made by the celebrated firm of Smith & Coventry. Thus I was equipped for cut-

ting threads on this system with extreme accuracy, in pitch, in form of thread, and in diameter.

6 The Sellers thread was an important improvement in three respects. It avoided the sharp edges at the top and bottom of the American thread, which were most objectionable. It increased the diameter of the bolt at the bottom of the thread by the aggregate height of four of the triangles which were cut off from the top and bottom, and it divided the acute angle of 60 deg. into two obtuse angles of 120 deg. each.

7 The Sellers improvement did not affect the pitches of the thread. These remained as originally fixed by Mr. Whitworth. Now for the first time the question of the real merit of the Whitworth thread in this latter respect presents itself, and it is found that it has no merit; that it is radically objectionable and ought to be abolished, for two reasons. First, it cuts into the bolt twice as deeply as is necessary. sacrificing unnecessarily about 20 per cent of its strength, and, secondly, it is inclined twice as much as it ought to be, making it impossible to tighten the nut properly, and rendering it easy for this to jar loose when exposed to even slight vibration.

8 In the years between 1890 and 1900, I projected a considerable advance in the high-speed reciprocating engine, and devoted much time to the study of plans for this engine. These, owing to the appearance of the turbine, have been laid on the shelf, where they seem likely to remain. Notwithstanding what I believe to be their very great advantages, it is certain that the psychological moment for taking them down from the shelf has not yet arrived—perhaps it never will. My reason for referring to this project in this connection is, that I became very deeply impressed with the utter unfitness, for the reasons above stated, of the coarse Whitworth threads for use in these engines. I spent much time in studying a system of finer threads by which these glaring defects inherent in the coarse threads would be avoided.

9 This system of finer threads, I presented in a brief paper to the Society at the December meeting, 1902. The proposed system covered the whole ground of the Whitworth threads from the $\frac{1}{4}$-in. bolt to the 6-in. bolt. In the former I made only a slight change, increasing the number of threads per inch from 20 to 24. In the 6-in. bolt the change made was more radical, increasing the number of threads per inch from $2\frac{1}{2}$ to 6. These extreme points were united by a series of nine steps, omitting the odd numbers of threads to the inch. I am inclined to think now that this omission was not called for, and I

will recommend a series of 18 steps, of which 8 are between the $\frac{1}{4}$-in. bolt and the 1-in. bolt, and the remaining 10 are symmetrically distributed between the 1-in. bolt and the 6-in. bolt. It will be observed that no change was made in the thread itself. I only cut smaller Sellers threads on larger bolts.

10 At the 1-in. bolt, the number of threads in the proposed system is 16 to the inch, just twice the present number. This affords an excellent opportunity for comparing the proposed system with the present one. Let us suppose a line to be drawn through the middle of the present thread from end to end. This line then becomes the bottom line of the proposed thread. The portion of the present thread inside this median line will be found to be worse than useless. Being nearer to the axis, its hold is more feeble; also for the same reason, its inclination is greater. It is the first to arrest tightening, and the first to let go. For these reasons its abolishment would be an important gain. That portion of the thread outside the median line will have only one-half the inclination and will be double in its extent; thus the ability of the nut to hold fast under vibration will be increased 400 per cent, from the fact that it has twice the holding surface with $\frac{1}{2}$ the inclination, but in addition to this, we are rid of that half of the thread inside the median line, with its disadvantages already mentioned.

11 It has been objected to this thread that it is not adapted to cast-iron and steel. I consider this objection to be a mere dictum without any foundation in fact. For three-quarters of a century the $\frac{3}{8}$-in. bolt has had 16 threads to the inch. Did any one ever hear of the objection to the $\frac{3}{8}$-in. bolt, that it was not suited for cast-iron and steel? I voice the universal answer, never! This bolt is perfectly adapted for use on cast-iron and steel; but if 16 threads to the inch are so adapted on the $\frac{3}{8}$-in. bolt, then they will be equally well adapted on any bolt on which we may wish to cut them.

12 It is objected again that these fine threads will strip. The answer is, that they will not strip. On the contrary, they are stronger than the coarse threads to resist stripping, for the reason that the circle on which the shear must take place is of larger diameter.

13 Finally, it is objected that they will wear away easily. On this point I have had a remarkable opportunity in my stone-dressing machine to show that wear is a question of material and not of surface. This experience is fully described in my Engineering Reminiscences. The bolts by which the tools were held in the tool holders were of wrought-iron, while the tool holders themselves were of steel.

These bolts, $\frac{5}{8}$ in. in diameter, were loosened and tightened 90 times every day through two years of service. At the end of that time, these bolts were as good fits as they were at the beginning, the reason being that the molecules of iron and steel do not interlock, while those of steel on steel do interlock; as shown in the case of the same tool holders, the adjacent surfaces of which were very soon torn to rags, while the end surfaces running in contact with cast-iron were only polished, without sensible reduction of either the steel or the cast-iron. This freedom of the different materials from wear is accentuated by the fact that no oil was used on these bolts, none being allowed where it might accidentally drip on the stone. The only lubricant of these bolts was stone-dust. The present system of using iron nuts on steel bolts, therefore, secures immunity from wear, irrespective of the extent of surface; but the surfaces in contact are also larger in the finer threads.

14 The responsibility is now before the Society of preserving the great feature of uniformity. It is obvious that this duty will not be performed by adhering to a system which is already being abandoned. Another chaos is now upon us. The automobile manufacturers have adopted a finer thread, almost identical with the one proposed by me, extending from the $\frac{1}{4}$-in. bolt to the 1-in. bolt, which is the largest bolt employed by them. The English engineers have become alive to the need of finer screw threads and have set about supplying this need in a very timid fashion. For example, they have increased the number of threads for the 1-in. bolt from 8 to 10, giving an increase in the strength of the bolt and an improvement in the security of the nut which is so small as to be ridiculous.

15 Mr. Burlingame proposes a medium increase in the number of threads, and he lands in the same hole with the English, 10 threads instead of 8 for a 1-in. bolt, where the system proposed by him stops. Two of our great tool builders have adopted threads of their own, which are being largely copied.

16 Mr. Trowbridge gives the whole case away in the following words, "Finer threads are undoubtedly very good for use where excessive vibration is encountered, but are not suitable for general use." This in effect recommends two systems; one adapted to resist vibration and the other not. This suggests a big hole for the old cat, and a little hole for the kittens. But it defies the fundamental law of machine construction, "*Always* provide for the extreme case, we cannot know when it may arise." That finer threads are not suitable for general use means only that we are not yet accustomed to them.

17 When the railroads learn the advantages of finer threads, they will not be strewing lost nuts and broken bolts over their roadways much longer, and will not keep men employed perpetually in tightening fishplate nuts.

The finer screw thread system is before the world. This Society may take the lead in its adoption.

THE AUTHOR The proposition of Mr. Burlingame to establish a new standard thread does not appear attractive. Actually it matters very little whether a single formula is used to cover the range of threads or more than one. For some years the great majority of manufacturing concerns have been using the U. S. standard thread for most of their work. To introduce a new standard to replace the old would cause a great deal of confusion and expense. The best solution of the matter seems to be the adoption of an entirely separate thread for sizes ¼ in. in diameter and over, called the U. S. fine thread. This could be made enough finer than the present standard to meet the demands of the automobile manufacturers and others who cannot use that, and yet be coarse enough to serve for the machine-tool builders.

2 In view of the action taken by the Society in adopting the motion of Mr. Halsey, that a committee be appointed to consider the question of a standard for fine threads, it seems desirable to leave to that committee the consideration of the exact standard to be adopted.

3 Mr. Porter suggests an extreme departure from the present "established" standard thread. He evidently intends to overlook the remark of Mr. Whitworth, who says: "But for this feature he would never have succeeded in procuring its adoption."

4 The "feature" was that Whitworth's proposed thread was only slightly different from the thread then in use. We can safely start any attempt to better the standard thread with one assumption in mind; *i.e.*, the U. S. standard thread is now in almost universal use in this country and will undoubtedly continue so for many years. Starting from this basis, we can see only one possible hope of getting finer standard threads and that is, the adoption of a U. S. fine thread.

5 Mr. Porter advocates a very radical fine thread and his arguments in favor of it are good. He does not tell us how to get the manufacturers to use it, however: so we had better try for the two systems, even if it does appear to provide "a big hole for the old cat and a little hole for the kittens." The kittens grow, provided they are healthy and well fed, and perhaps the fine thread will have as happy a future.

No. 1201

THE BY-PRODUCT COKE OVEN

By William Hutton Blauvelt, Syracuse, N. Y.
Member of the Society

Coke is the product of the dry distillation of bituminous coal, or in other words, coke is the solid residue remaining after bituminous coal has been heated in a closed, or partially closed, chamber until all of the volatile matter has been driven off. This volatile matter consists mainly of tarry matter, gas and ammonia, familiarly known as by-products. The quality and quantity of the coke and the by-products vary considerably with the quality of the coal. In this country coals similar to those from the Pocahontas region, containing from 16 to 18 per cent of volatile, stand at one end of the list, and produce the maximum yield of coke and the minimum yield of by-products. On the continent of Europe some coals are coked containing not more than 13 per cent of volatile matter. At the other end of the list stand the highly gaseous coals, which contain as much as 38 or 40 per cent of volatile matter and yield correspondingly small amounts of coke.

2 Good coke has a clearly defined cellular structure, with hard cell walls, the cells being small in size, and comprising about 55 per cent of the total volume of the coke, so that its bulk for a given weight is about twice as great as that of broken anthracite coal. The well developed cellular structure and the hard cell-walls are important in the use of coke for metallurgical purposes, and account for its superiority in blast furnace work, for example, over anthracite coal, which was formerly the principal blast furnace fuel. The cellular structure presents a large surface to the oxidation of the air, producing rapid combustion, while the firm cell-wall prevents crushing and maintains an open fuel bed. The specific gravity of a good quality of standard coke is about 1.7 to 1.9.

3 Many investigators have attempted to discover the substance

Presented at the Detroit Meeting (June 1908) of the American Society of Mechanical Engineers.

which makes coal coke, with a view to determining why some coals will coke and others will not. A French chemist, Mr. Lemoine, believes that he has isolated this substance and gives it the name of "carbene." It is a black solid friable material tending toward crystallization, having the formula $C_{22}H_{14}O_5$. Experience has demonstrated that coals which are high in oxygen are usually deficient in coking qualities, and either will not coke at all, or produce a coke having a weak, friable structure. As will be shown later, many coals which produce very indifferent coke under ordinary conditions, by proper treatment can be made to yield a good metallurgical coke.

4 The production of coke in this country has grown very rapidly during the last decade, as is indicated by the following figures. The column showing the production from by-product ovens indicates that the growth of this branch of the industry has been fairly rapid, although several conditions have united to check it during the last few years.

TABLE 1 PRODUCTION OF COKE, NET TONS

	Total	From by-product ovens	Percentage of by-product coke
1897	13 288 984	261 912	1.97
1898	16 047 209	294 445	1.87
1899	19 668 569	906 534	4.61
1900	20 533 348	1 075 727	5.24
1901	21 795 883	1 179 900	5.41
1902	25 401 730	1 403 588	5.53
1903	25 274 281	1 882 394	7.45
1904	23 661 106	2 608 229	11.02
1905	32 231 129	3 462 348	10.74
1906	36 401 217	4 558 127	12.52

In the above figures coke from gas works is not included.

5 A little study shows at once that the several types of coke ovens have sprung from three "root forms," the beehive oven, developed from the mound of the charcoal burners; the Coppée oven, with its vertical side heating flues, and long narrow coking chamber; and the Knab-Carvés, with the same narrow coking chamber and with horizontal side flues.

6 In the beehive oven the coal is coked in a flat layer, say 12 feet in diameter, and about 24 to 26 inches thick. In the retort oven the coal lies in a long narrow chamber from 30 to 40 feet long, from 6 feet to 8 feet high, and from 16 inches to 22 inches wide. The essential distinction between the beehive and retort oven processes is that

in the former the combustion which produces the heat for coking the coal, takes place within the coking chamber, above the coal. Air is admitted at the front door, and the products of combustion escape through a hole in the roof. In the retort oven the heat is generated by the combustion of gas in flues in the side walls: the heat passes through the walls into the coking chamber, but the products of combustion and the air are carefully excluded from the coking chamber. In the latter process, therefore, there is a true distillation of the coal, while in the former (the beehive oven) the admission of air materially changes the composition of the tars so that they are richer in the oils and paraffines, more like the tar produced from the Scottish blast furnaces, for example, where raw coal is used in place of coke. The admission of the air reduces the value of the tar products, as well as the quantity, and also destroys practically all the ammonia. The tar from the closed, or retort oven, on the other hand, contains considerable quantities of benzene, naphthalene, and anthracene, and other of the aromatic hydrocarbons which have given coal tar its great value in the manufacture of so many organic compounds, dyes, drugs, etc.

7 The retort oven may be arranged to recover the by-products from the distillation of the coal, or not, depending upon the market conditions and the quality of the coal coked. The coking process is essentially the same in both cases, except that when the by-products are not recovered the gas from the coal is led directly to the oven flues, while, when they are to be saved, the gas coming from the ovens is cooled and scrubbed to free it from the tar and ammonia, and is then returned to the oven flues. In both cases regulation is, of course, necessary, so that a suitable amount of gas is admitted to the flues to carry on properly the coking process.

8 The author has given in some detail, in Mineral Industries, volume 4, page 215, the various steps of the development of by-product ovens from the earliest types of the beehive oven, and the various forms in use in Europe. As our interest is rather in the by-product oven as it exists today in this country, this discussion will be confined to the types which have representatives here.

9 The beehive oven is, in general, familiar. Fig. 1 shows its ordinary form. As now built it is about 12 feet in diameter inside, and about 6 feet 9 inches high to the top of the domed roof. The ovens are built in single or in double rows, banked together with masonry to prevent loss of heat. A charge of about six tons of coal is delivered by larry through a hole in the roof. Each charge of coal

is fired from the heat remaining in the walls from the preceding charge; the air for combustion enters at the front door, the products of combustion escaping through the charging hole in the roof, producing the "pillar of cloud by day and pillar of fire by night" familiar to travelers in our coke regions. When the operation is complete, the coke is quenched with water by means of a pipe introduced through the front door, and after cooling, the coke is drawn, either by hand or by machine, to the wharf in front of the ovens, whence it is loaded into cars. The labor of drawing the ovens is the most expensive and laborious item connected with the operation, and much study has been given to devices for drawing the coke. A machine

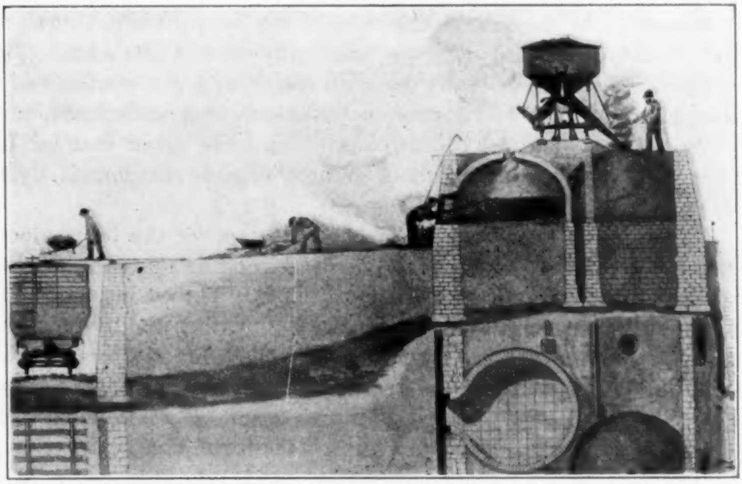


Fig. 1 Section of Beehive Oven

recently developed, which promises permanent success, is shown in Fig. 2. It will be noted that it is arranged to travel on a track in front of the ovens, and when the oven is open and the arm thrust in, the coke is pulled out on the conveyor shown in the illustration, which automatically delivers almost the entire contents of the oven directly into the railroad car.

10 The coking process occupies from 48 to 72 hours, so that the output of a beehive oven is from 1.5 to 2 tons of coke per day. The United States Geological report for 1905 gives 394.8 tons as the average output per active oven in that year. The beehive oven has been carried to its highest perfection in this country, and the above

mentioned drawing and loading apparatus is only one among the comparatively recent improvements.

11 The use of silica brick in the construction of the beehive oven has become quite general, and is a distinct step in advance, notwithstanding that the general experience with silica brick, in regard to its rapid destruction when suddenly cooled, caused nearly all coke manufacturers to be very skeptical regarding its use until it had actualy been tried.

12 lIn the present condition of our iron industry, subject as it is to extreme fluctuations, the beehive oven has certain advantages. It is quickly built and at relatively low cost, and the labor required for

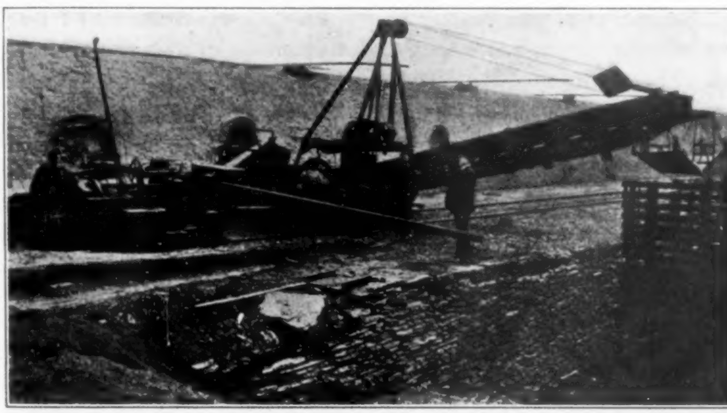


Fig. 2 Coke Drawer and Loader for Beehive Ovens

operation is of low grade; it may be put out of run at relatively small loss during the periods of industrial depression, and it can be started up again with ease after a shutdown. But with changed conditions the iron industry is becoming more stable; and moreover, the coals which were especially adapted for use in the beehive oven are becoming more and more scarce, especially with the approaching exhaustion of the Connellsville field. There are very large deposits of coal in this country which are not well suited to the beehive oven, but which under proper treatment will make excellent coke in the retort oven. In many sections of the country these deposits are the natural source of supply for important metallurgical plants. In this respect the case is similar to that in some important iron districts of continental Europe, where the available coals may be coked only in

the narrow retort oven, and where the beehive oven has long since entirely disappeared. It is especially in such cases that our attention is turned to the retort or by-product oven. This is essentially a long narrow firebrick chamber, from 30 to 40 feet long, from 7 to 9 feet high, and from 16 to 20 inches wide. It is closed at each end by doors lined with firebrick and usually there are from three to six holes in the roof through which the coal is charged. In the side walls and under the bottom of the chamber there are heating flues in which is burned a portion of the gas produced by the distillation of the coal, the heat passing through the firebrick walls into the coal. These heating flues are arranged sometimes vertically and sometimes horizontally. The ovens are built in blocks of from 25 to 50 or more. The distillation of the coal in this tightly closed chamber by heat from without, forms the essential characteristic of the retort oven.

13 The products of the distillation of the coal pass from the oven chamber through a hole in the roof to a large pipe, or main, extending along above the block. In some designs this main contains some water and the gas is forced through it in a manner similar to the hydraulic main of the gas works. The distillate, as it comes from the ovens, consists of a gas similar in general composition to ordinary illuminating gas, and it is loaded down with water and tar vapors and also carries ammonia. If the tar and ammonia are to be recovered from this gas, it is led to the by-product recovery building for treatment. If not, a proper proportion of the gas is returned to the oven flues and burned there with a properly regulated supply of air. The remainder, if any, is available for raising steam. The air is usually preheated either by regenerators, similar in principle to those of the Siemens furnace, or by transmission of the heat through thin walls of firebrick, commonly called recuperation. The heat for preheating the air is taken from the hot gases leaving the oven flues. In many plants these hot gases, after giving up a portion of their heat to the air, are passed under boilers for raising steam for operating the plant.

14 In the plants which recover the by-products, the gas coming from the main is treated in apparatus similar in general to that used in the manufacture of illuminating gas. The temperature is reduced gradually, so as to avoid the shock of the gas with a consequent precipitation of naphthalene and the lighter hydrocarbons which form the principal illuminants of the gas, and is finally brought down to about the temperature of the atmosphere. This work is done in large vessels cooled either by air or by water. There are a number of types of these coolers or condensers, a common form being a series of vertical

cylinders containing tubes like a vertical boiler. Through these tubes water is circulated which cools the gas progressively as it passes from one vessel to another. During this cooling process most of the tarry matter and water vapor are thrown down, the latter carrying with it considerable quantities of ammonia.

15 After these condensers come the tar extractor and then the scrubbers, whose function is to remove the remainder of the ammonia and tarry vapor, and sometimes also the benzol and cyanides. The gas is drawn from the ovens and forced through these various apparatus by means of a rotary exhauster, operated either by a steam engine or electric motor. The ammonia is removed in a scrubber which may be of the "bubbling" or of the "rotary" type.

16 The "bubbling" type consists essentially of a vertical cylinder containing a series of compartments having inverted cups or bells with serrated edges dipping into water, so arranged that the gas bubbles through the water in fine streams, passing upward through the several compartments. Water is fed in at the top and runs out at the bottom as ammonia liquor containing from one-half to two per cent of ammonia. The rotary scrubbers are essentially large horizontal cylinders with a shaft running through them horizontally. On this shaft are arranged metal sheets, or brushes, or wooden slats so disposed as to offer the maximum amount of surface to the gas as it passes through the several compartments into which the cylinder is divided. The slow revolution of the shaft keeps the surfaces wet, the lower part dipping into water which partially fills the vessel. Water is fed in at one end and the liquor runs out at the other in the same manner as in the other apparatus.

17 The benzol and cyanide scrubbers are similar in construction, but instead of water, heavy oil of tar or petroleum is used to absorb the benzol, and for the cyanides, ferrous sulphate in an alkaline solution. After this treatment the gas is ready to be returned to the ovens for heating the flues, or to be sent out for use in gas engines, for illuminating gas, or for fuel, as market conditions determine. If it is to be used for illuminating purposes, it must be purified of its sulphur compounds in a similar manner to illuminating gas. When destined for this use it is also desirable to retain all of the illuminants. The removal of benzol is therefore omitted, and frequently advantage is taken of the fact that in the distillation of coal the early portion of the distillation produces gas much higher in illuminants than the later portion. The gas piping from the ovens is therefore arranged, under these circumstances, so that the first portion of the gas coming off from each oven,

say during the first six or eight hours of the coking process, shall be kept separate from the remaining portion, lower in illuminants. In some plants the benzol is recovered from the portion carrying the lower illuminants, which is to be used for heating the ovens, and when this benzol is added to the richer portion, a gas is produced of much higher candle power than could be obtained from the same coal in the ordinary gas works, for example.

18 The tar and the ammonia liquor recovered from the various operations just described are collected in large tanks, and separated by the difference in their specific gravities. The tar is pumped to tanks for shipment, and the ammonia is sent to the ammonia distilling plant. This ammonia liquor, commonly called "weak liquor," contains, as above, from one-half to two per cent of ammonia, also some of the lighter tar oils. The ammonia exists partly in the form of sulphate, chlorid and other fixed salts, from which it can be freed only by the admixture of lime, or some similar alkali, and the remainder exists as "free ammonia" in combination with carbonic acid, hydrogen sulphid, etc. When all of the fixed ammonia has been freed from the acid by the addition of lime, it is driven off from the liquor by boiling with steam, the condensed gases carrying a certain portion of water, to form the crude ammonia liquor of commerce. By the introduction of additional apparatus and the modification of the process of distillation, so as to separate the ammonia gas from the accompanying impurities, the aqua ammonia of commerce may be produced. Or the ammonia gas coming from the distillation apparatus may be absorbed in sulphuric acid, producing sulphate of ammonia, which is described more at length elsewhere.

19 The recovery of benzol from the heavy oil used as an absorbent is effected by heating with steam in apparatus similar, in a general way, to that used for the ammonia. The benzol oils are driven off and condensed as crude benzol ready for purification, and the heavy oil is returned to the scrubber to be used again.

20 The modern by-product oven plant includes, among its more important mechanical problems, the delivery of the coal to the ovens and the removal of coke. The earlier installations in this country were small and the quantities of coal and coke handled did not justify the installation of much special apparatus, but with larger installations, consuming from 1500 to 3000 tons of coal per day, and producing an equivalent quantity of coke, the handling apparatus has become one of the important features of nearly every plant. The location of these plants at the blast furnace, and the absolute depend-

ence of the furnace on the coke produced, has made it necessary to store coal to prevent interruption of the operation from strikes, car shortages, etc. Also the location of some plants demands the storage, during the winter months, of coal which is brought by vessel in the summer. Then it is necessary to install unloading machinery. For example, an oven plant designed under the writer's direction a short time ago, is arranged with a capacity for unloading over 400 tons of coal per hour from vessels, and with apparatus for receiving this coal from the vessel and storing it at the same rate up to an amount of 300 000 tons. This apparatus will, at the same time, reclaim coal from this storage pile at the rate of 200 tons per hour and deliver it to the coal bins.

21 At this plant, among others, the engineering problem was further complicated by the necessity of mixing two coals of different composition in order to produce a coke of suitable quality. So the apparatus had to be arranged to store these coals separately, reclaim them separately, and bring them together in a mixture of proportions determined by the quality of the coals, and deliver this mixture to the oven bins. It was also necessary to grind the coal, in order to insure the best physical structure of the coke. The standard at this plant is that over 70 per cent must pass through a one-eighth inch mesh screen, and this work must be done at the rate of at least 200 tons per hour. Such undertakings present some very interesting problems in mechanical engineering, and upon their solution depends to a great extent the economical operation of the whole plant.

22 The operation of charging the ovens is a very simple one. The coal, prepared as described above, is delivered in bins above the ovens and drawn into a larry or car which runs over the top of the oven blocks, operated by a motor. This car is arranged with suitable chutes, and on opening the charging holes on the tops of the ovens the coal is allowed to run in and fill the oven chamber. These charging holes are then immediately closed, and the top of the coal is leveled off by means of a reciprocating ram, operating through one of the doors at the end of the oven. This leveling produces sufficient space for the gases to escape and reach the exit pipe, and also to provide for the swelling of the charge which usually takes place during the coking.

23 When the gases have been driven off from the coal, which requires a period of from 18 to 30 hours, depending upon the coal and the oven, the doors at each end of the oven are opened, and the mass of coke is pushed out by a ram, operated by steam or electricity.

This ram travels up and down in front of the ovens as required, and usually carries the leveling device above mentioned. By this method the coke is discharged in a few minutes, and the oven is ready for another charge, which is dropped in as soon as the doors are closed.

24 The coke, on discharging, must be quenched. The old way, and the one largely employed in Europe still, is to spread the hot coke on a brick pavement and water it with a hose. This does not usually give the best results, however, as the coke is blackened and frequently is made too wet. A number of devices have been developed for quenching, with varied success. In one, the coke is run into a tight firebrick chamber and water is admitted, the pressure of the steam generated driving the water throughout the different parts of the mass. In another plan, the coke is pushed out into a car 40 feet long and 10 feet wide, having an inclined bottom so arranged that by drawing the car along as the coke is pushed out it is distributed in a uniform layer, and is then quenched by a heavy stream of water, which is applied as soon as the coke reaches the car. Another device is in the form of a box-shaped frame, built up of water pipes so arranged that the coke is pushed out through it and quenched by a shower of water forced through perforations in the pipes—a sort of needle bath.

25 After quenching, the coke is either delivered into railroad cars or carried directly from the original quenching car to the blast furnace bins, or to a sorting and picking plant, in case the product is to be sold for foundry coke. In this picking plant the coke is delivered from a small hopper on the picking belt, where the black ends are removed, as is the custom in preparing foundry coke, and the small pieces are screened out, the coke being delivered to the car without any handling except of the rejected pieces.

26 All of this work was formerly done by hand, and the difficult quality of the material and the severe requirements of the market required the development of some special apparatus in order to accomplish the results mechanically, with low repairs and good efficiency.

27 With this general use of machinery, of course, the power plant becomes an important part of the installation, since, in some installtions, at least, all the apparatus and machinery are electrically driven.

28 Consumption of power in a plant of by-product coke ovens, of course, depends very much on the arrangement of the plant, and especially upon the conditions under which the coal and coke have

to be handled. The following is the distribution of power consumed at one plant coking about 1500 tons of coal per day. The figures represent the average daily consumption in kilowatt hours. Of course, many of the operations, such as pushing and charging the ovens, are very intermittent.

	kw-hr.
Handling coal to and from storage pile	450
Coal conveyors	900
Crushing and pulverizing coal	1130
Gas exhausters	1280
Rotary scrubbers and pumps in by-product recovery plant	1250
Pumps handling ammonia liquor	255
Pushers	684
Coal charging	168
Coke handling	1440
Pumping water	1080
Lighting	730
	9367

29 Some special problems in design arise on account of the large requirements of steam for the distillation of the ammonia and other by-products. On this account a simple type of engine, operating under (say) 10 to 15 pounds back pressure, presents strong arguments for favor in comparison with more efficient engines. The use of the exhaust steam for distillation makes available the latent heat in the steam, as well as the sensible heat down to the temperatures at which the waste liquors are discharged. So that while by this method only 20 per cent of the heat in the steam is consumed in the engine, the remaining 80 per cent becomes available for the work of distillation. The steam consumption in such an engine is about 45 pounds per horse power. But when the exhaust steam is needed for distillation it is hard to equal the economies of this combination as compared with other types of steam engines, or even gas engines, combined with the use of live steam for distillation.

30 In addition to the problems of handling materials and the development of the mechanical appliances about the plant, which have been special features of the growth of the by-product oven in America, much progress has also been made in the oven itself. Although this retains its original form, the size and coking capacity have been much increased. The earliest retort ovens in this country had a cubical capacity of 220 cubic feet, and had an average coking capacity of 4.4 tons of coal per 24 hours. Some of the ovens of today

have a capacity of 430 cubic feet, or more, and coke about twelve tons of coal per day. Experimental installations having an output materially above this figure are operating satisfactorily.

31 Among the mechanical devices connected with the retort oven is the stamping machine. The illustration in Fig. 3 represents one

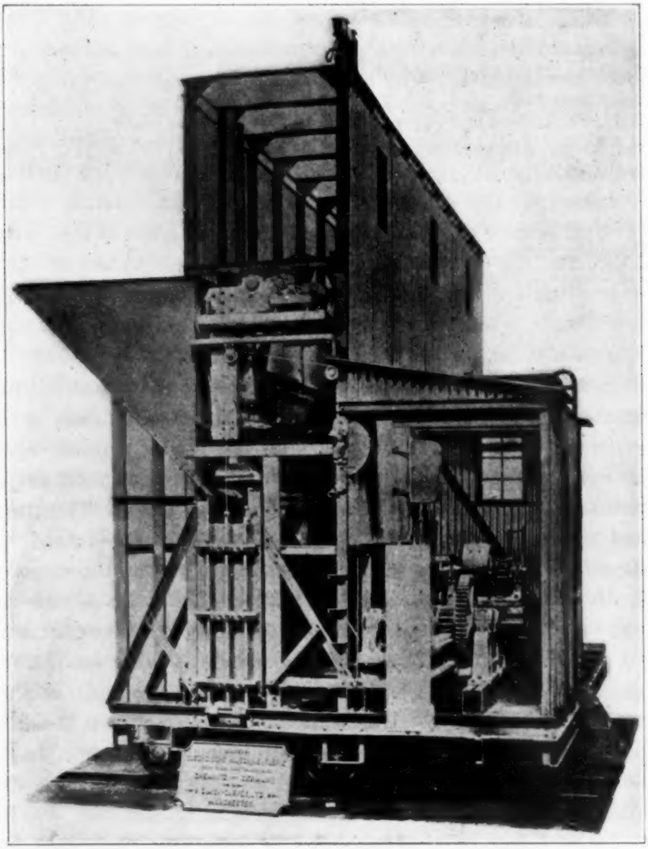


FIG. 3 COMBINED COKE STAMPER AND COKE PUSHER

of the successful German machines of this type. It has been used largely in Europe, but only to a small extent in America, for the purpose of compressing the coal into a cake before charging into the oven. Many coals, on account of their composition, do not produce a coke of satisfactory physical structure by the ordinary methods of

treatment, but when they are compressed into a cake, the physical structure of the coke is very much improved.

32 The process is in principle a simple one. A box or form is provided, slightly smaller than the oven chamber. This box is filled with coal, and during the process of filling the coal is compressed by heavy stamps traveling back and forth the length of the box, so that there is a reduction of volume of from 20 to 25 per cent, as compared with the loosely charged coal. The coal is dampened before stamping, and it is not difficult to form a cake having sufficient strength to permit of being pushed into the oven. Besides the improved quality of the coke, the output of the ovens is at many plants increased by about 10 per cent, and in some cases the managers state that they prefer this method of charging coal into the oven, even aside from its effect on the quality of the coke. The illustration shows a pusher combined with the stamping machine, so that the whole apparatus travels up and down together in front of the ovens, the coal being stamped in transit.

33 Five different types of by-product recovery ovens have been built, or are building, in this country. One of these, the Newton Chambers, is an English adaptation of by-product recovery to the beehive oven, but the plant which is the only American representative of this type of oven has been out of run for some time. This system consisted essentially of a series of vertical cast iron U-pipes, arranged to receive the gases from the beehive oven and condense out the tar and ammonia vapors, by cooling with air or water, the scrubbed gas being returned to the oven, which was heated partly by the flues underneath, and partly by combustion in the upper part of the chamber containing the coal. As pointed out before, this method cannot but give low yields of ammonia and tar of inferior quality.

34 The four types of by-product ovens in operation and under construction in the United States are all of the retort oven type, and have all been developed from the original types of the Coppée or Knab-Carvés designs. Of these there are 11 plants of the Otto-Hoffman and the United Otto type, 13 plants of the Semet-Solvay type, 3 plants of Rothberg ovens, and 1 plant of Koppers ovens, making a total of 28 plants or 4763 ovens. Detailed descriptions of these ovens having appeared elsewhere, I shall confine myself to a statement of the distinguishing points of the several systems.

35 The accompanying illustration, Fig. 4, shows a typical United Otto oven plant in cross section. The oven is supported on steel girders resting on concrete piers. It is of the vertical flue, or Coppée

type; that is, the gas is burned in a series of vertical flues arranged in the side walls between the oven chambers. The gas for heating the flues, which has been scrubbed and cooled in the by-product plant nearby, is delivered into the main combustion flue, both at the end of the oven and below in the gas pipes shown. The products of combustion pass into one of the regenerative chambers, and thence to the stack, and the air is preheated in these chambers, on the principle of the familiar Siemens regenerative furnace. The currents of gases and hot air are reversed at suitable intervals. During each of these periods the gases pass upward through one half of the oven

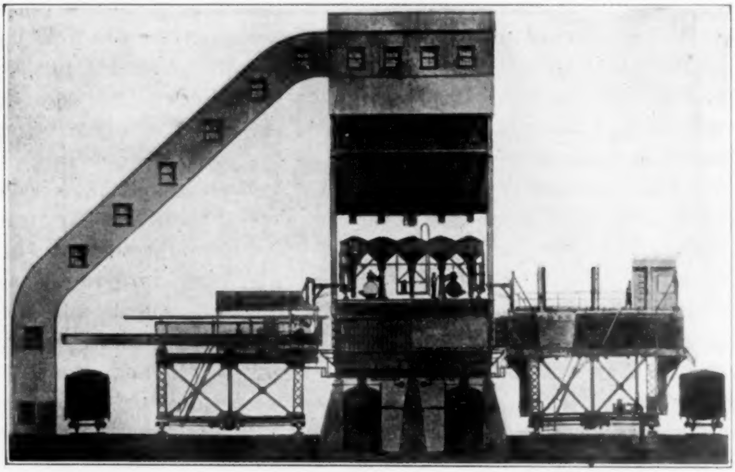


FIG. 4 SECTION OF UNITED OTTO OVEN

flue system, and downward through the other half, their path being reversed at each reversal of the regenerator valves.

36 The Semet-Solvay oven shown in Fig. 5 is a development of the Carvés type. Its salient feature is the horizontal flue system with strong middle walls which support the roof structure and the additional loads of charging larry, etc. The horizontal flue system is heated by means of gas introduced into the ends of all but the lowest flue. At each admission it meets with air for combustion which has been preheated by the escaping gases through the thin walls of the "recuperator" arranged below the oven. The horizontal flues with the multiple gas admission admit of very accurate control of the heat and of its even distribution throughout the coking cham-

ber. The main object of the central division wall above mentioned is the prolonged life of the oven structure which results from taking the weight off the thin and highly heated flues, and it also permits repairs to be made to any oven without shutting down the adjacent ovens.

37 The Rothberg oven is of the horizontal flue type, but more like the Huessner oven which has been developed in Germany, in that is has only one flue system in each case between the coking chambers. This system is divided in the center by a vertical wall, forming two compartments with short horizontal travel for the gas. Every alternate flue is served by a gas burner, and all flues can be inspected by walking along the platforms at the ends of the oven. A series of

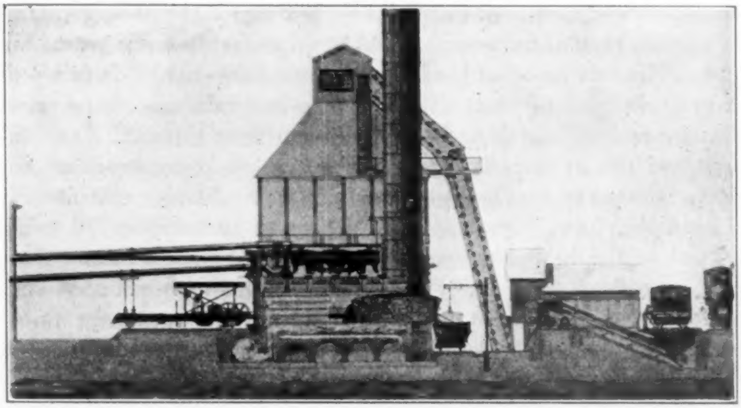


FIG. 5 SECTION OF SEMET-SOLVAY OVEN

dampers in the flues is arranged, which cause the gas either to travel the whole length of the flue system, or to pass directly to the next lower flue, in case the heating conditions of the oven demand it. These ovens may be built either with or without recuperators or regenerators.

38 The Koppers oven is of German development, and the first plant is now under construction in this country. It is a vertical flue oven of the Coppée type. Its essential features, as described in the American patents, are the special arrangements for the control of gas and air and their combustion in the oven flues. These consist essentially of the arrangement of the ports, so that the gas and air travel spirally through the flues, thereby carrying the heat higher up in the oven and preventing local combustion at the point of entrance;

also a control of the effective area of each flue, so as to distribute the heat uniformly over the side of the oven chamber, and provision for examining all the flues through openings arranged for that purpose. The air, and sometimes the gas, is preheated in regenerators arranged underneath, one pair of regenerators for each oven.

39 In the by-product apparatus used in the various oven plants and in the general arrangement for handling the coal, coke, and other products, there are many variations, but these are rather to meet the conditions of the individual locations than inherent in any special oven system.

40 In addition to the various problems in mechanical and chemical engineering presented by the operation of the by-product coke oven, there are a number of metallurgical problems, some of which have been solved with material gain in the efficiency of the coking operation and the yield of by-products; the solution for others is yet to be found. The driving off of the hydrocarbons from coal by distillation is a very complex process. Each change in conditions of the temperature, rate of distillation, etc., gives different results, and on account of the hydrocarbons being so complex in composition, no definite figures can be placed on the operation. Many experiments, for example, have been made to determine the amount of heat absorbed in the coking process per unit of coal coked. Many processes have been devised for converting the major portion of the nitrogen in the coal into ammonia. These and other problems, however, are not much nearer solution than when the by-product process was first introduced into this country. The experience in the retort house of the gas works has shown that high heats and rapid coking increase the volume of the gas and tend to break down the heavier hydrocarbons, and the same is doubtless true in a measure in the retort oven, although the much greater bulk of the charge, the different relation between the volume of the gas produced in proportion to the heated surface of the retort, etc., modify these conditions.

41 The coking time of a charge in the oven depends upon the temperature of the heating flues, and the rate of transmission of the heat through the wall. Until recently "quartzite" brick has been used for these flue walls, in accordance with the European practice. Quartzite is the ordinary firebrick with sufficient quartz rock added to make the brick about neutral in its expansion and contraction, and contains about 60 to 70 per cent silica. Recently, however, silica brick has been introduced somewhat generally, notwithstanding the additional complication of its considerable expansion under the

temperatures of the oven chamber. Silica has the advantage of not being injured by the higher heats required by latest practice, and is also a materially better conductor of heat than the quartzite. Magnesite is very attractive on account of its very high conductivity, but this material has not yet been successfully used.

42 The conductivity of firebrick increases with the temperature. There are no data available giving the exact relation of conductivity of either clay or silica brick at coking temperatures, but experience shows that a moderate increase in the thickness of the flue wall does not necessitate a materially higher temperature in the flue in order to maintain a given temperature in the coking chamber.

43 It has not been observed that the yield of coke from any given coal is affected by the speed of coking. When the coking operation is properly conducted, with entire exclusion of air, there is always deposition of hydrocarbons, and an increased yield of coke above the theoretical. This increase in yield is probably largely due to the pressures existing in the oven chamber, and the prolonged period of contact of the gases with the coal in process of coking. The retort oven always yields higher percentages of coke than can be obtained either in the crucible or in the gas retort, probably for the above reasons. The yields of by-products are not affected by temperature to anything like the same extent in the retort oven as in the gas retort where it is recognized that the higher temperatures produce larger volumes of gas higher in hydrogen and lower in hydrocarbons, together with lower yields of tar. As suggested above, these phenomena are doubtless due to the different relations between volume of retort and volume of charge.

44 The presence of water in the coal when charged increases the coking time, since, of course, the water must be driven off before the coking process begins, and therefore the penetration of the heat into the charge is delayed by that amount. The addition of water up to 10 or 12 per cent does not add seriously to the coking period, but the amount of heat absorbed is considerably increased, and there is also additional work thrown upon the condensing apparatus, in cooling the products of distillation.

45 The rate of distillation of the coal in the oven (known in practice as the coking time) varies with the quality of the coal and with the content of volatile matter; but this rate does not vary directly with the percentage of volatile matter, probably because in the coals containing the most volatile matter distillation proceeds at a much more rapid rate during the early part of the process.

This is illustrated by the accompanying table, showing the amount and composition of the gas produced from a low volatile and a high volatile coal during each hour of the coking process.

46 The coke from the by-product or retort oven differs in some respects from that made in the beehive. On account of its being quenched outside of the oven it has lost the silvery glaze seen on the

TABLE 2 ANALYSES AND YIELD OF GAS FROM LOW VOLATILE COAL

Hour	Cu. ft. per hr.	CO_2	C_6H_6	C_2H_4	Total Ill.	O	CO	CH_4	H	N	B. t. u. per cu. ft.
1	536	0.1	0.5	6.0	6.5	0.8	4.6	31.6	42.1	14.3	643
2	425	0.1	0.7	3.6	4.3	0.8	4.3	32.8	51.6	6.1	638
3	475	0.1	1.1	3.8	4.9	0.7	4.9	33.2	46.8	9.4	641
4	357	0.1	0.9	3.3	4.2	1.1	4.6	33.5	49.6	6.9	637
5	411	0.1	0.8	3.7	4.5	0.8	4.6	33.1	50.8	6.1	644
6	401	No sa	mple	taken							
7	517	0.2	0.7	2.5	3.2	1.0	4.4	30.1	44.4	16.7	560
8	387	0.2	0.5	2.9	3.4	0.9	4.5	32.6	46.2	12.2	598
9	400										
10	320	0.0	0.4	1.6	2.0	0.9	4.1	29.1	47.4	16.5	
11	256										
12	189	0.1	0.5	1.2	1.7	0.6	4.6	29.5	53.6	13.9	
13	251										
14	256	0.1	0.2	0.6	0.8	0.9	4.4	17.0	69.2	17.6	
15	387	0.0	0.3	3.5	3.8	4.9	2.9	3.0	44.0	41.4	
16	425										
17	260										
18	37										
19	17										
20	0										
					Unpurified						
4		1.9	1.1	4.0	5.1	0.8	4.6	33.0	46.7	7.9	
6		2.0	1.2	3.3	4.5	0.9	4.4	30.0	44.7	10.0	
8		2.2	0.8	2.8	3.6	0.8	4.2	31.2	46.2	11.8	

Total cubic feet..6307

beehive coke. Formerly this was thought to be of some importance, as the glaze was supposed to help resist the action of carbonic acid on the coke in the blast furnace. Repeated experiments, however, have shown that the retort coke is rather more resistant to carbonic acid than beehive coke made from the same coal. Owing to the difference in method of applying the heat the shape of the pieces of coke is different. This difference in structure is shown in Fig. 6. In the beehive oven the heat generated by the combustion of the gases above

the coal works downward, developing a long finger-like structure. In the retort oven the heat is applied from both sides and works toward the middle, resulting in short blocky pieces somewhat less in length than half the width of the oven. Retort oven coke is somewhat denser than beehive coke from the same coal, the percentage of cells being slightly smaller.

47 There is still some difference of opinion among furnacemen as to the relative value of the two cokes in the blast furnace, owing to

TABLE 3 ANALYSES AND YIELD OF GAS FROM HIGH VOLATILE COAL

Hour	Cu. ft. per hr.	CO_2	C_6H_6	C_2H_4	Total Ill.	O	CO	CH_4	H	N	B. t. u. per cu. ft.
1	829	0.9	0.9	3.2	4.1	0.5	5.8	41.5	41.4	5.8	693.4
2	562	0.9	1.0	2.6	3.6	0.4	5.1	40.4	43.8	5.8	681.5
3	582	0.9	1.0	2.1	3.1	0.7	4.9	37.6	47.2	5.6	654.2
4	582	1.1	1.1	2.1	3.2	0.4	5.0	36.2	48.6	5.5	648.5
5	582	0.8	1.0	1.7	2.7	1.0	4.6	33.3	49.5	8.1	588.6
6	452	1.1	0.9	1.6	2.5	1.1	4.6	31.4	49.8	9.5	583.8
7	490	2.2	0.9	1.3	2.2	1.6	4.4	31.0	47.6	11.0	566.2
8	449	0.9	1.0	1.5	2.5	0.5	4.8	31.5	54.2	5.6	603.0
9	432										
10	456	1.0	0.5	2.0	2.5	0.6	4.9	29.1	55.3	6.6	
11	474										
12	470	0.6	0.1	0.3	0.4	0.4	5.3	23.1	64.8	5.4	
13	494										
14	405	0.5	0.0	0.4	0.4	0.6	5.3	18.2	67.0	8.0	
15	460										
16	460	0.2		0.0	0.0	0.4	6.2	13.6	69.4	10.2	
17	386										
18	388										
19	148										
20	34										
21	48										
22	13										

Total cubic feet..9194

differences in individual experiences. Retort oven coke has received a great deal of criticism in this country because much coke has been made from coal which would be quite out of the question for use in the beehive oven. The reasons for using such coals were doubtless satisfactory to the management of the oven plants, but the coke produced has been often held up for comparison with Connellsville beehive coke, which is perhaps the very best beehive product in the world. There are very many different grades of retort and beehive coke produced from the different coals, and under different conditions

in the several plants in operation, that a definite comparison of the two general types of coke is practically an impossibility. Some of the best furnace work in the country has been done with retort coke, which would seem to justify the statement that the best retort coke is as good as the best beehive coke in the blast furnace.

48 In the foundry cupola retort oven coke has an exceptional record, and in some localities it is without question the standard of foundry coke.

49 In the beehive operation some of the fixed carbon is burned, while in the retort oven no combustion of the carbon can take place, and the yield is, therefore, up to or above the theoretical. On this

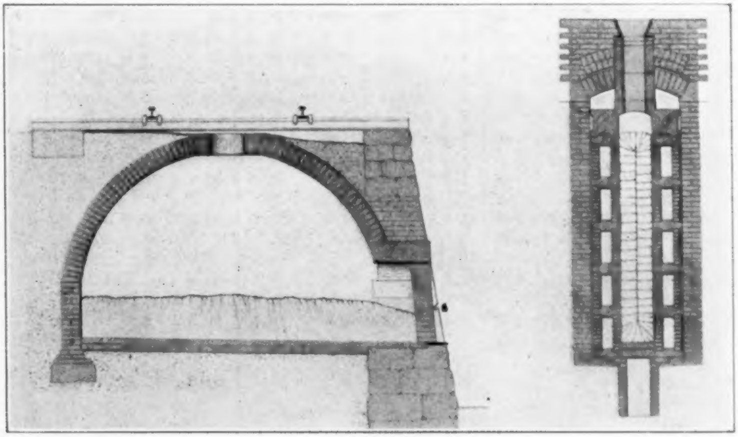


FIG. 6 STRUCTURE OF COKE IN BEEHIVE AND RETORT OVEN

account the retort coke is always somewhat purer chemically than beehive coke from the same coal. This gives the former a somewhat higher percentage of carbon, and lower percentage of ash, phosphorus, etc.

50 The by-products produced at plants operated in this country are tar, ammonia, gas and benzol. Cyanides are not being recovered at any of the American plants.

51 The tar is the standard coal tar of commerce, and does not differ materially from that produced by gas works. In Europe the great briquetting industry is built up on the basis of tar from by-product ovens, since coal tar pitch is the only binder for briquettes which has been largely successful. Pitch constitutes from 60 to 70

per cent of the tar, the remainder being heavy oil, consisting of creosote oil, napthalene, anthracene, etc., and the light oils which are mainly impure benzols. These are separated by distillation, the pitch remaining as residue. As is well known, coal tar is the basis of a large number of important industries, especially in continental Europe, but only a portion of the tar is available for these higher grade products. Anthracene is the raw material for alizarin, and napthalene is the basis of artificial indigo. Carbolic acid is formed from the creosote oils, and from the benzols are derived most of the long list of aniline colors, the manufacture of which has become such an important German industry.

52 The practical absence of these chemical industries in this country has been one of the causes of the relatively slow growth of the by-product oven here. While it may be some time before the manufacture of these refined products obtains an important foothold here, yet the briquetting industry, after many years of experimenting and failure, is at last becoming established commercially. Some half dozen plants located in different parts of the country have their product on the market on a commercial basis, and their growth should be rapid.

53 The low prices and high quality of coals that have been generally available, together with high labor cost, have delayed the establishment of this industry here, but the increasing price of anthracite has encouraged the development of briquettes from the smaller and unmarketable sizes of anthracite from the coal yards and washeries in the East. In the Middle West the mining of some of the noncoking coals is accompanied by the production of a large amount of fine coal not available for general use. This and other relatively low priced coals form a considerable supply for briquetting. Farther west the lignites and semi-lignites offer a very promising field. Many of these coals break down badly upon exposure to the atmosphere into a fine powder which is practically useless. Briquetting prevents this disintegration, and a properly made briquette will stand a year's exposure to the elements without apparent injury, while the coal from which it is made would be reduced to a powder within a week or two. The engineers in charge of the Fuel Testing Station of the United States Government at St. Louis have made an interesting series of tests covering the briquetting of a large variety of coals, and their results are most encouraging, especially on the classes of fuels last mentioned.

54 The process of briquetting is essentially a simple one, although

the conditions necessary for success with each fuel must be definitely determined and maintained. The plants are not expensive or complicated, and now that works are operating commercially under widely different conditions and environments, a satisfactory growth of the industry should no longer be delayed.

55 The increased price of lumber and the destruction of our forests is bringing the preservation of lumber by creosoting prominently to the front, especially for railroad ties, and the demand for creosoting oil is already increasing. New uses are being found for the benzol oils, as will be shown below.

56 Moreover, the application of tar to road surfaces is rapidly growing in favor. Especially since the introduction of the automobile it is found that no material produces a better road surface for the money expended than tar. This is sprinkled on the surface of the road and rubbed in with brushes, and if a penetration of an inch or so is effected, the surface of the road is impervious to water, and the presence of the tar produces an elastic condition of the surface which has a rather remarkable effect in increasing the wearing qualities of even indifferent road-making material. Tar can be quickly applied either to a new or an old surface, preferably the former. Dust is eliminated almost entirely, and by resprinkling every year or two, depending upon conditions of travel, material, etc., the road surface is kept in perfect condition, except, of course, for the replacement of material actually worn away.

57 By-product ovens have for some years been the leading producers of ammonia in this country. The total value of ammonia produced by gas works and by-product ovens together, in the year 1907, is given by the United States Geological Survey as $4 126 529. Figures are not available showing the output of the by-product ovens alone.

58 This ammonia is sold in the form of crude ammonia liquor, sulphate of ammonia, and aqua ammonia. The first named is an impure solution of the ammonia gas in water, containing some light tar oils, sulphur compounds, carbonates, etc., and contains from 15 to 25 per cent of ammonia. It is used in the ammonia soda industry, and also as a raw material for manufacturing higher grades of ammonia such as the nitrate and carbonate of ammonia, etc.

59 Sulphate of ammonia, a grayish white crystal containing about 25 per cent of ammonia, is used as a fertilizer, and somewhat for the manufacture of anhydrous ammonia. In Europe the use of sulphate of ammonia as a fertilizer has been very greatly developed,

partly by active educational methods, bringing home to the farmer the great gains from its use. It divides with nitrate of soda the responsibility for the great sugar-beet crops of continental Europe, and a great deal of study has been given to the intelligent application of this most available form of nitrogen to the varying crop conditions there found. In this country the development of the use of sulphate of ammonia as a fertilizer has only begun. While figures are not available showing the amounts of ammonia consumed in all the various uses, yet the following table will be of interest, showing a comparison of the ammonia productions of the several leading countries in the year 1906. This table is from a paper by Mr. C. G. Atwater, published in Mineral Industry. The figures represent all ammonia products figured as sulphate.

	metric tons
England	279 000
Germany	235 000
United States	68 000
France	49 100
Belgium and Holland	30 000
Other European countries	55 000
Total	716 100

60 When we consider that the production of sulphate in this country was only 25 000 tons and that about 23 000 tons were imported during the year, the meager consumption of sulphate in this, the greatest of agricultural countries, is very striking. While, of course, the newer farming lands are still unimpoverished and do not require fertilizing, yet in all the older states crop yields have fallen off very greatly during the past generation, and yet the present use of artificial fertilizers is mainly confined to the Southern states on the cotton and tobacco fields. A consideration of our acreage of tilled land compared with that of Germany, for instance, will show the possibilities of the use of sulphate when our agricultural colleges and government agricultural stations have educated the farmers up to a full realization of the importance of this invaluable plant food.

61 Aqua ammonia, which is essentially a purified solution of ammonia gas and water, containing about 25 per cent ammonia, is the refined "ammonia" of commerce, usually purchased in a very diluted form. During recent years its use has grown very rapidly for refrigerating purposes, both as aqua ammonia and as the source of

anhydrous ammonia. The latter product is made essentially by driving the ammonia gas off from the water by heat, drying it and condensing it into a liquid by pressure.

62 The surplus gas from the by-product oven is the portion remaining after sufficient gas has been used for heating the ovens, and the amount varies greatly with the coal used. In lean coals, low in volatile matter, there might perhaps be no surplus, while in rich gassy coals the amount may be from 4000 to 5000 feet per net ton of coal. As stated above, this gas is essentially similar to that made in gas works. Following is a typical analysis:

	per cent
Carbon dioxid	1.3
Benzene	1.2
Ethylene	4.2
Oyxgen	0.5
Carbon monoxid	5.1
Methane	35.5
Hydrogen	48.0
Nitrogen	4.2
B. t. u. per cubic foot, gross	679

63 If the gas is desired for sale for illuminating purposes and the richer portion is reserved for this purpose, as described above, the gas is richer in marsh gases and illuminants and contains less carbonic oxygen and hydrogen than the average. The calorific value of the gas may vary from 550 to 700 B. t. u. per cubic foot. In this country its use for illuminating purposes has been developed much more than in Europe. There are now eleven plants supplying illuminating gas regularly for city consumption, and in 1907, 17 885 400 000 feet were sold, compared with 36 934 300 000 from coal gas works. In Europe the surplus gas has been largely consumed in gas engines. At one time there was some trouble on account of pre-ignition of the gas, which was laid to the high percentage of hydrogen. This has been overcome, however, and leading engineers no longer consider it an obstacle to its use for this purpose. It has been shown by Mr. Pennock in the Journal of the Society of Chemical Industry, June 15, 1905, that when coke oven gas is mixed with the proper amount of air for combustion, the mixture has not as high a percentage of hydrogen as producer gas, for example, which is largely used in engines without trouble from this cause.

64 Benzol is recovered from coke oven gas very largely in Europe, and is the raw material for the synthetic manufacture of the anilines and of carbolic acid, as well as a number of other chemical products.

It forms the principal illuminant in coal gas, and its use is growing in this country as a convenient source of candle power, especially in small gas works. It is also used in the manufacture of varnishes.

65 Benzol has been used experimentally to a considerable extent both in Europe and in this country in explosion engines, both alone and mixed with alcohol. A series of carefully conducted tests in this country on stationary engines, motor boats and motor cars has brought out several points in its favor as a fuel for such engines. An equal volume has about 12 per cent more calorific value than ordinary gasolene. Owing perhaps to the fact that benzol is distilled within closer limits than gasolene, it has a somewhat higher thermal efficiency, and is not so liable to miss fire in the cylinder.

66 After this general review of the subject the question remains—What is to be the development of the by-product oven in this country? Compared with many industries its growth has been slow, but this, as has been pointed out, is largely due to the condition of the chemical industries in this country, which must of necessity keep pace, at least in a measure, with the sources of their raw materials. The capital required in the installation of the by-product oven is relatively large, and the results would have been serious if the rate of oven building had been greatly in excess of the demand for its by-products. In all industries depending upon the by-products of coke a promising future is apparent. The use of gas for illuminating purposes has become firmly established, and the rapid introduction of the gas engine affords a market for fuel gas. The various uses of ammonia in the arts have already been discussed. But we must look to agriculture as the principal consumer of any largely increased production of ammonia. Through the growing influence of Government and State experimental stations and agricultural colleges, science is being applied more and more to agriculture, and this cannot but lead to the general use of artificial fertilizers which naturally follows a higher standard of farm work. As a fertilizer base, sulphate of ammonia, with its nitrogen in the most available form, plays a most important part.

67 During the last few years the production of tar has exceeded the demand, so that large quantities have had to be burned as fuel at low prices, but the outlook for this by-product is more favorable than it has been. While it may be some time before the highly specialized chemical industries for the production of anilines and dyes have reached any important growth, yet the prospects for the development of the cruder tar products are encouraging. The

rapidly increasing price of lumber is forcing the necessity of creosoting, with the attendant consumption of creosoting oil, on large lumber users like the railroad companies.

68 The practical start in the commercial manufacture of briquettes which has been described, promises an early development of that industry, and a very modest position as a producer of fuel would mean consumption of a large amount of pitch. If the number of by-product ovens and the output of tar therefrom were doubled, a production of less than 3 000 000 tons of briquettes per annum would consume all the pitch from this increase, without providing for any growth in the important roofing and paving industries, and without providing for any consumption of tar in the new tarred roads. This tonnage of briquettes, compared with the 86 000 000 tons of anthracite and the 388 000 000 tons of bituminous coal mined in 1907, suggests that briquetting, when once established, even as an "infant industry," may soon reach the condition which exists in continental Europe, namely, where its growth would depend only upon the available supply of pitch. Briquetting pitch must always be sold at a moderate price, unless the conditions of our fuel supply change materially. But, as in Europe, when once developed, briquetting will probably be the principal consumer of the pitch produced in this country.

69 The by-product oven belongs to the present day. The production of coke at the blast furnace plant, where the operation is assured by large storage of coal and where the labor is under better control than in less developed regions, is in line with today's manufacturing methods. The higher economy of its operation, its special adaptation to control by modern scientific organization, the ability to draw coals from several fields, and the successful adaptation of labor-saving appliances to its various operations, all point to its being the oven of the future.

DISCUSSION

Mr. J. R. Bibbins It is fortunate that this subject of by-product coke ovens has been brought before the Society by Mr. Blauvelt at this time, especially with the promise of a demonstration upon so large a scale. The by-product coke oven offers so many interesting phases of commercial development that it is difficult for the layman to understand the reasons for its somewhat limited present application. Yet

a more careful study of the operation of the large by-product coke plant soon reveals the necessity for the coördination of many important factors, any one of which might operate to disturb the balance of the whole.

2 To the outsider the most interesting phase of coke manufacture is the production of by-products. The Detroit installation serves as an illustration of the use of by-product gas for illuminating or power purposes. It is the popular impression that the sale of by-product gas for this purpose represents clear profit without any difficulties or embarrassments encountered in the guarantee of an uninterrupted supply of gas. Although there are many factors to be considered, it is true that under certain conditions power generation from by-product coke gas is commercially profitable, both to the supply company and to the customer. At a reasonable price for gas, power may be generated at a cost per kilowatt so far below the cost of steam power as to preclude the use of steam entirely.

3 In this connection some data regarding the first coke oven gas installation of large size in this country may be of interest. This development has taken place at Lebanon, Pa., where two large by-product coke plants are in operation, one of the Semet-Solvay and the other of the United Otto type. Until about a year ago there was no opportunity to dispose of the by-product gas from either of these plants, and all the surplus was allowed to go to waste. In 1907, two power projects were developed—one to operate the works of the American Iron and Steel Company, and the other to establish a large gas engine plant near the ovens to furnish power for the ovens and mines supplying them, and to transmit a large amount of surplus power to the city of Lebanon for public utilities. The first of these plants was installed in 1907 and has been in continuous operation since, from gas piped several miles under moderate pressure from the Semet-Solvay coke plant. The second project will be actively pushed forward within the next few months. Both employ horizontal double-acting engines of the twin-tandem type of 500 and 1200 h.p. capacity respectively.

4 At the time of this first Lebanon installation very little was definitely known concerning the possible success or failure of this kind of gas, as the only precedents in this country were a small vertical single-acting engine at Syracuse and a larger engine at Camden, N. J., which was in commission for several years, operating on poor gas from the last half of the coking run.

5 The results of a year's operation at Lebanon have been ex-

tremely satisfactory. Trouble anticipated from excessive hydrogen content has not developed, and after some experimenting for sulphur removal, no serious trouble has resulted from the somewhat high sulphur content. At the outset, the builders stipulated a maximum content in the gas of 50 per cent hydrogen by volume and 0.02 gr. of sulphur per cubic foot. Both these limits have been exceeded without resulting trouble. Although pooled gas (i.e., run-of-oven) is supplied, the hydrogen content at times runs as high as 66 per cent by volume, and the sulphur $1\frac{1}{2}$ gr. per cubic foot.

6 This sulphur exists in the form of H_2S and CS_2 in varying proportions. The gas is purified in rectangular oxid purifiers containing iron oxid sponge prepared on the ground. By giving proper attention to this apparatus and renewing one layer of oxid each week, the sulphur may easily be reduced to 80 or 85 per cent of the original content, which runs from $2\frac{1}{2}$ to 4 gr. per cubic foot in the crude gas. This leaves about 0.6 gr. sulphur in the gas supply to the engine, much more than was originally considered safe.

7 As to the effect of sulphur upon the engine, it apparently passes through the combustion cycle without difficulty, provided three simple remedies are employed:

a To insure absolute separation of the water and gas around the cylinder valves or packing;

b To flush the rods thoroughly with continuous streams of engine oil;

c To maintain the piston cooling water circuits at a fairly high temperature, sufficient to prevent condensation and the consequent formation of acid.

8 In all modern designs, a continuous return lubricating system is employed. It is a simple matter to provide a small stream of oil in front of each packing cage, which returns to the filter and is again pumped through the system without loss.

9 A fairly high piston water temperature is advantageous, not only from the sulphur standpoint, but to facilitate lubrication as well. In fact most of the packing trouble experienced in the early days of the horizontal design was found to be due to running the rods too cold, as a matter of precaution.

10 During the early experience of the Lebanon plant, the sulphur reduction decreased for some time to a point as low as 50 per cent, or 2.0 gr. per cubic foot at the end of the week. Recently a measurement of the rod diameters showed that the reduction during the year had been extremely small, and at one packing could hardly be detected.

This seems to indicate that in a proper design and with necessary precautions the sulphur problem is not so serious as anticipated. Recently this engine was operated six hours on crude unpurified gas containing over 3 gr. of sulphur, and has been so operated at intervals during the past year. With normal lubrication the rods tended to run dry in places, which might cause considerable trouble at the packing, but by flushing the rods with a small stream of oil, this trouble was immediately corrected.

11 One important phase of the Lebanon situation is the relative value to the coke oven operator of principal and by-products, and herein lies the crux of the application of coke oven gas for power purposes. The power user of course desires a guarantee of uniformity of quality and continuous supply; and this necessitates the operation of the ovens for the purpose of delivering the by-product even in periods of decreased demand for coke, tying up the capital of the operator in stored coke, to his serious embarrassment. At Lebanon the gas is supplied by a contract that fixes the rate, but not the supply, which is dependent on the operation of the coke oven plant. This provides cheap power and heat for the consumer, and relieves the coke oven operator of the above objection. On the other hand, this arrangement would be entirely unsatisfactory to a consumer contracting for a definite service unless he took the precaution of installing a relay producer equipment. Such a plant has already been installed at Lebanon to provide for a contingency of this nature, and at the prevailing low price of producer apparatus it would certainly be cheaper in the long run than to employ steam power, providing the coke oven gas could be obtained at a reasonably low figure in comparison with coal. This whole question therefore resolves itself into an economic one which both producer and consumer must solve.

12 Another point of interest in regard to the Lebanon installation is that it is an alternating current system with generators solid coupled, and operating at a frequency of 40 cycles per second. The unit operates regularly in parallel with an adjoining steam plant containing both tandem and cross-compound steam engines. At this intermediate frequency no trouble whatever has appeared in parallel operation.

Mr. C. M. Barber Mr. Blauvelt's paper is certainly a very valuable contribution to our knowledge of coke manufacture. The writer would like, however, to advance a few ideas in regard to the *cooling of coke.*

2 That the oven heat of coke from by-product ovens can be removed without the use of water is a proposition that the writer has demonstrated to his own satisfaction. The proof is based on the fact that the largest single pieces that are usually discharged from the Semet-Solvay oven, if isolated from contact with the mass, will cool without combustion in the open air.

3 A charge of good coke when pushed from the oven while it has a high temperature is not undergoing combustion. The term "quenching" implies combustion. We would substitute the word "cooling" on the ground that the coke when ready for discharge from the oven has parted with all those components of the coal which would ignite when exposed to the air. The coke itself, if properly handled, does not ignite. The process therefore, it seems to us, is one of cooling rather than of quenching.

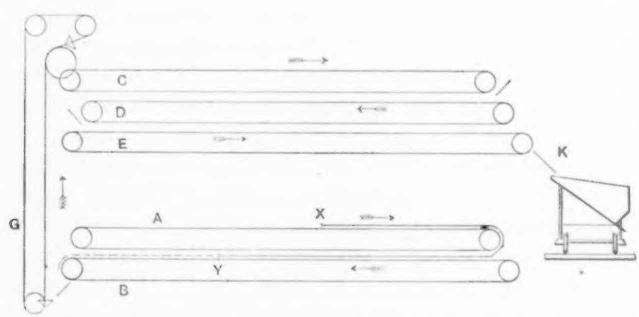


FIG 1 ARRANGEMENT OF CONVEYORS FOR RECEIVING AND DISCHARGING COKE

4 If the mass is simply spread out so that the largest pieces practically lie by themselves, the coke will give up its heat just as a newly rolled bar of steel cools in the open air.

5 In discharging the coke from one oven directly upon a conveyor, say, 6 ft. wide by 150 ft. long, the average thickness would be less than 3 in. Considering a block of 30 ovens and allowing 22 hours for coking, an oven is discharged on an average of every 45 minutes. Five such conveyors could be arranged to give a cooling period of about three hours, which would simply mean that the coke is detained about three hours on its trip from the ovens to the furnace.

6 In Fig. 1, *A*, *B*, *C*, *D* and *E* are conveyors, each about 150 ft. long by 6 ft. wide. They are adapted to receive the coke from a block of 30 Semet-Solvay ovens. *A* and *B* displace the ordinary

coke car and its track. The top of A is about one foot below the floor of the oven and receives the charge of any oven of the block. C, D and E may be located directly over A and B. G is an elevator having buckets adapted to receive the coke from B and discharge it on to C. A and B are coupled to work together at the same speed, which is variable and controlled.

7 The operation is as follows: The contents of an oven are slowly discharged by the coke pusher in the usual way onto the conveyor A, at any point X. The man operating the conveyors regulates the speed of A and B so as to distribute the entire contents of the oven over about 150 ft. of conveyors A and B.

8 This carries the charge to about the point marked Y on conveyor B. He then continues the movement at any speed he wishes until the point Y has arrived at the discharge end of conveyor B, when he locks his lever at a speed just sufficient to discharge the entire contents of the one oven from conveyor B on elevator G in about 45 minutes, which will leave conveyors A and B empty when it is time to use them for the next oven.

9 The operation already described will require from about three to five minutes, or the usual time of discharging an oven plus the time for completely covering conveyor B, making the whole operation not more than an average of about six minutes.

10 Elevator G and conveyors C, D and E are all run at one uniform speed of about 150 ft. in 45 minutes, and the coke is finally discharged at K into a car or in some cases directly into the bins at the blast furnace.

11 The above is a general description, showing the lines upon which the apparatus can be worked out.

12 The advantages of cooling coke from the oven temperatures to that required for handling without the use of water are almost too apparent to require noting here. We may, however, call attention to the effects of dashing upon the incandescent coke quantities of water sufficient to cool it to blackness in a few seconds. Coke has a cellular structure. The cell walls are hard and vitreous. Struck with a hammer, good coke will often ring like crockery. It contracts considerably on cooling. The almost instantaneous cooling cracks and breaks it very much as the same treatment would break almost any like substance. We know that glass, furnace slag, cast iron and many other substances fly to pieces when suddenly cooled with water. The watering of incandescent coke is always accompanied by cracking, snapping sounds caused by the disintegration of the coke. We

know too that a considerable amount of the breeze is caused by the effect of the water.

13 The delivery of coke to the furnace bin with a low content of moisture is the aim of the coke maker, but the difficulty of getting the water to the interior of the pile as it lies on the car is considerable. In fact no water reaches the inner part until the outside is overwatered and the ideal condition of leaving just sufficient heat to completely dry the coke before the car reaches the bins it is practically impossible to realize.

14 We believe that it will be conceded that coke cooled without water is harder and stronger, and will stand dropping into bins with less breakage. The quantity of breeze will be less, and since the coke is in larger pieces and not full of shrinkage cracks, there will be less of it lost by the dissolving action of the carbon dioxid gas in the upper zones of the furnace.

15 The increased value of the coke will far more than doubly repay the probable cost of 750 ft. of conveyors moving at a speed of 3 ft. per minute.

Mr. C. G. Atwater The discussion which I have to offer on Mr. Blauvelt's complete and well considered summary of the by-product coke oven situation up to date, consists of the operating figures of the industry in this country for the year 1907, which naturally were not available at the time the paper was written. These figures have been compiled from reports made to me by all, or nearly all, of the oven operators. The courtesy of these gentlemen, among whom is Mr. Blauvelt, I wish to acknowledge here.

2 The total figures are given in the following table:

OPERATING DATA FOR BY-PRODUCT COKE OVENS IN U. S. FOR THE YEAR 1907

Coal carbonized, net tons	7 391 285
Coke produced, net tons	5 492 425
Tar produced, U. S. gallons	53 393 495
Ammonium sulphate, or sulphate equivalent, produced, net tons	62 700
Gas produced and sold, cu. ft. (from reports and estimates)	12 588 000 000
Total gas produced beyond that used for oven heating, cubic feet estimated too heavy	15 500 000 000

3 These items I will discuss in order: The coal carbonized in 1906 in by-product ovens, as given by the U. S. Geological Survey, was 6 192 068 net tons, or something over a million tons less than was carbonized in 1907. Similarly, the coke they produced in 1906 was

4 558 127 net tons, very close to a million tons of coke behind the 1907 figures. The yield of coke per ton of coal was 73.6 per cent in 1906, and 74.3 per cent in 1907, this difference being probably due to the increased carbonization of low volatile coal yielding a high coke percentage, in the latter year.

4 The total coke production of the United States was but three or four hundred thousand tons greater in 1907 than in 1906, as nearly as preliminary estimates can ascertain; therefore it may be seen from the previous statement that the gain in total coke production was all due to the by-product oven, and that the production from beehive ovens actually fell off to some extent. Up to the last two months of the year the coke production from both sources promised to be ahead of any previous year, but the industrial depression intervened, and the beehive ovens, being better adapted to shutting down and starting up again on short notice, naturally suffered first. If we assume that the total production was 37 000 000 net tons, which is probably beyond the actual figure, the proportion of by-product coke produced is very close to 15 per cent. For the year 1906, as shown by the figures given in Mr. Blauvelt's paper, the proportion was 12.5 per cent, so that the year 1907 will probably show a good advance here, as was to be expected from the other figures. If we review the figures for years past, as given in the paper above-mentioned, we find that such an advance from year to year has occurred several times, and that the by-product coke oven has made steady progress since its introduction.

5 The production of tar in by-product ovens in 1905 is given as 36 379 854 gal., the report on this product, as well as that on gas and ammonia, having been omitted for the year 1906 by the Geological Survey. The figure given in the table for 1907 shows an increase of something over 17 000 000 gal., or about 47 per cent for the two years. The actual yield of tar per net ton of coal carbonized was 7.22 gal. in 1907, 7.86 gal. in 1905 and 7.77 gal. in 1904. This decrease in tar per ton of coal is due to the increasing use of the lower volatile coals, to which cause the increased yield of coke was also ascribed. Though these coals give lesser yields of gas and ammonia, as well as tar, they give an excellent quality of coke as well as a larger quantity, so that their use will probably increase as time goes on.

6 The production of ammonia, in the form of ammonium sulphate or its equivalent, amounted to a little less than 62 700 net tons, or about 17 lb. of sulphate per net ton of coal carbonized. When it is

considered that this figure covers all the vicissitudes of operation throughout the whole industry, it must be regarded as a very creditable one indeed, though doubtless still capable of improvement.

7 The figure for gas produced and sold cannot be regarded as strictly accurate, as some of the items included in it are estimated. It is intended to represent the amount of gas from by-product ovens that actually came upon the market, on the same basis as city gas or natural gas. Some of it was used for heating steel furnaces or glass furnaces or for steam making, or for direct power generation in the gas engine, but a good proportion of it was used in general city distribution, either alone or in combination with carburetted water gas. It does not include gas from all the plants, however, as some are operated on low volatile coal which produces only enough gas to heat the ovens themselves properly, and at other plants the gas is not sold at all, but is used for raising steam or for other purposes around the plant, or wasted. The amount of coal carbonized at the plants reporting their gas as all or partly sold was about 5 600 000 net tons, or there was about 2250 cu. ft. of gas disposed of per net ton of coal carbonized. It is believed that this figure is, if anything, below the actual facts.

8 The figure for total gas produced beyond that needed in heating the ovens is intended to include the gas used for steam raising or other purposes around the coke plants themselves, wasted for lack of an immediate use or by leakage in distribution, or which for any other reason cannot be included in the figure for gas sold. The amount is estimated in many instances, but such estimates are made on a very conservative basis.

9 To turn to another phase of the by-product coke oven subject, viz., the use of coke-oven gas in the gas engine, I would like to ask a question. It has been stated to me, as a point developed in experience with a coke-oven gas containing considerable sulphur, that the piston rod of the engine lost its lubrication and became dry when the cooling water in the rod was allowed to get too cold, but that if the water was kept up to a moderately high temperature, the oil adhered better and the danger of scoring the rod appeared to be less. Such an action might be attributed to more condensation of acidulated moisture on the cooler rod, hence more removal of the oil than when the moisture and acid vapors remained volatile and went off in the exhaust. I would like to know whether any one has had experience with sulphurous gases that bears upon this point, and whether the explanation suggested is in accordance with our knowledge of the

somewhat obscure reactions that take place in the cylinder of an internal combustion engine.

PROF. R. H. FERNALD Referring to the action of sulphur on gas engines, we operated the engines in the government testing station in St. Louis for 2½ years and used no purifier. We used bituminous coals, lignites and peats. The highest percentage of sulphur in any fuel used was 8.2, contained in California lignites. These fuels were used without cleansing as far as the sulphur was concerned and we had no difficulty at all with the engines so long as we carried out the scheme of Mr. Bibbins for keeping the water away from the sulphur. We carried the gas directly into the engine and discarded the purifier. When the plant was removed we left the purifier in St. Louis and have made no use of it since.

MR. JOHN C. PARKER Two points brought out by Mr. Bibbins' very interesting discussion and his citation of the Lebanon plant present themselves to the speaker as being of especial interest.

2 Mr. Bibbins has referred to the sale for power purposes of such by-product gas as may occur and to the use of this by the customer as it may be generated. I should like to inquire what source of supply is used during the period when the coke ovens have not a sufficient supply of by-product gas to furnish the necessary demand of the gas engines. Is it economically possible for the customers to carry stand-by producer plants?

3 The other point involves the determination of price in the sale of any economic utility. Two things determine the sale price: the cost of production and the value of the utility rendered. In the case of by-product sale the cost of production is indeterminate between that of the fundamental product and the by-product, but the compound cost of production is very definite, depending upon the ratio upon which the fundamental product and by-product may be sold. If there is a large demand for one of the products, which, for the sake of discussion, we will call the fundamental, the price of this will tend to rise, and in order to dispose of the by-product the price of it must be marked by the value—in competition—of the utility rendered. If in the early days of the business this price is placed to meet competition it is likely to be set at a very low figure which may induce a comparatively large demand. It would be difficult, however, to raise the price, as in the sale of any public utility there is a general prejudice against the raising of price for any reason whatever. Where

by-product gas is sold in competition with other forms of power it might seem that there would be a danger of over sale resulting in forcing the main product to become a by-product, as has happened in so many industries. In the case of the sale of a public utility, such as gas for power purposes, is it right to refuse arbitrarily the sale of power gas before such a condition is reached; or should a different method be used to prevent the growth of a demand which may necessitate the manufacture of the fundamental product in quantities beyond the economic market value, thus running down the profit on the fundamental product?

Mr. G. J. Rathbun A manufacturer of lubricating oil stated to me that he was having trouble with lubrication of piston rods on a gas engine, where the gas was known to be high in sulphur. I advised him to run hot water through the rod instead of cold and he afterwards stated that this overcame the difficulty.

The Author In considering the use of coke oven gas for gas engines the two points usually considered, aside from the heat value of the gas, are the hydrogen and sulphur contents. In a gas in which the nitrogen is properly controlled the hydrogen may vary from 40 to 55 per cent. The higher percentages of hydrogen in coke oven gas, compared with other gases, were formerly thought a serious objection. This has been overcome, as Mr. Bibbins and Dr. Fernald have stated. From the reports of some German engineers I find that they are able to prevent pre-ignition entirely in their practice by varying the compression in inverse proportion to the value of the gas mixture. A proper control of the admission of air and gas by the governor, so that the largest charge and weakest mixture are introduced when running under light loads, and the smaller charge and more powerful mixture under heavy loads, prevents injurious pre-ignitions and produces an equally inflammable mixture and uniform explosion. By proper care on these points some of the German plants have operated on coke oven gas for periods of four months without cleaning or stopping the engines.

2 The sulphur in coke oven gas exists in two forms, namely hydrogen sulphid and carbon bi-sulphid. The former may be readily removed by the ordinary purification process, but the latter cannot be removed by ordinary methods. Fortunately it exists in much the smaller proportion. Of course the amount of sulphur in the gas depends upon the coal from which it is made. The stipulations of

the builders of the Lebanon plant, referred to by Mr. Bibbins, provided for a gas very low in sulphur. The ordinary specifications for purified illuminating gas stipulate a maximum of 20 gr. of sulphur compounds per 100 cu. ft., but this is of course a thoroughly purified gas. Coke oven gas made from coals suitable for metallurgical purposes contains before purification from 1 50 to 400 gr. per 100 cu. ft. 660 gr. per 100 cu. ft. equals one per cent by volume. Good German gas engine practice does not regard 0.2 per cent of sulphur as harmful and does not purify the gas even when the percentages run higher. They consider that copious lubrication of the parts exposed to the gas after combustion prevents any harm. If the large percentages of sulphur present in Professor Fernald's St. Louis experiments can be used without harm by the simple expedient of keeping the parts warm to prevent condensation, we may soon hope to see the sulphur question removed from consideration, and I think we may look forward with some confidence to the absence of apparatus for sulphur purification in a gas engine plant using coke oven gas.

3 In the consideration of coke oven gas for power purposes, the questions of supply and cost are of course essential. Perhaps these points may be made clearer by a brief consideration of the by-product oven operation. A by-product coke oven plant is so called because originally coke was the only product from a coke oven, and when by the improvement of the oven other products were saved, they were called by-products. At most plants coke is still the primary product, and especially where metallurgical coke is made the recovery of by-products is not allowed to injure the production of the best possible coke. In a retort oven plant, however, the coal is distilled for the purpose of recovering several products, which may include coke, tar, ammonia, benzol, cyanides, gas, and perhaps others, any one of which may be the primary product, and all the others by-products. In the distillation of coal in a gas works, for instance, the gas retort takes the place of the oven and the primary product is gas, and the coke, tar, ammonia, etc., are by-products. In the large plant of by-product coke ovens at Everett, Mass., it might be correctly said that gas is the primary product, and the coke, tar, and ammonia the by-products, since the gas is used to supply illuminating gas to a large number of consumers and is an important part of the gas supply of that vicinity, while the coke, to a considerable extent, is sold as domestic fuel and in whatever quantity may be produced in furnishing the necessary supply of gas.

4 In undertaking the installation of a by-product coke oven

plant, the owner must consider the operation as whole. He has certain products to dispose of in the available market, and he must make his contracts and arrange his operation to meet the demand and the prices obtainable. If he has contracted to deliver a certain product at the maximum rate of production, it may become necessary to produce more of some other product than can be disposed of to the best advantage. This would result in loss unless he were able to modify his operation, perhaps by using another coal, or by some improvement in process, so as to effect a re-adjustment of relations between the several products. The prospective owner of a plant considers fully the value he may expect from each of the products of the operation in connection with the cost of production, and the so-called "by-products" are no longer uncertain in quantity and value, and treated as a secondary consideration, though it may have been so to some extent in the earlier days of the by-product oven.

5 Of course the application of these statements to the use of coke oven gas in gas engines for power plants is that the operator of an oven plant contracting to deliver a certain amount of gas to a power plant would expect to live up to his contract to the same extent as if he had agreed to deliver a given quantity of tar, ammonia, or coke.

6 In order to guard against accidents beyond the control of the oven manager, and insure a continuous operation of the power plant, it might be advantageous to install a producer plant of sufficient capacity to operate the gas engine. As Mr. Bibbins has intimated in this discussion, the cost of such insurance would not be prohibitive. Or the producer plant may be installed at the coke ovens and the gas used to supplement the portion of the oven gas ordinarily consumed in heating the ovens. In practice the operation of the modern by-product coke oven plant has not been found subject to sudden or great fluctuations, and its products may be relied upon as fully as those from any manufacturing plant.

No. 1202

THE HORSE POWER, FRICTION LOSSES AND EFFICIENCIES OF GAS AND OIL ENGINES

By Prof. Lionel S. Marks, Cambridge, Mass.
Member of the Society

For a whole century the indicated horse power of the steam engine has been accepted as the most satisfactory measure of the work done in the cylinder of that engine. When the gas engine came into use it was but natural that the same measure of its power should be used. So long as the whole cycle took place in one cylinder there was but little doubt as to what was meant by the indicated horse power of the engine; but when auxiliary air and gas pumps were used, the indicated horse power required special definition. A committee of the Society reported in 1902 a code of rules which contained this special definition, and which has given a consistent meaning to the indicated horse power of steam engines and of gas and oil engines of all types.

2 The definition referred to is not, however, universally accepted. The pages of the *Zeitschrift des Vereins deutscher Ingenieure* for 1905 contain a long and most animated discussion by many of the ablest German engineers on the meaning of indicated horse power and mechanical efficiency in two-cycle engines, and they show very marked differences of opinion as to the correct method of calculating those quantities. Within the past few months the definition of the indicated horse power of a four-cycle engine has been the subject of debate by the British Institution of Mechanical Engineers, and a strong tendency has manifested itself to take as the measure of the indicated work of a four-cycle engine only that area which is included in the positive loop of the indicator card.

3 In all cases it has been assumed that the indicated horse power is the best measure of the work done in the engine, but the differences of opinion as to the methods of its measurement are really indications

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

of the fact that the indicated horse power of an engine, and the quantities deduced from it, do not give that information which engineers have been trying to extract from them. The fundamental trouble with the indicated horse power as the unit of power is that it does not represent the actual work done by the working substance, but the difference between that quantity and certain resistance. Consequently it does not permit a comparison to be made between the actual amounts of work done by the working substances in the cylinders of engines of different types.

4 So far as the steam engine is concerned, the indicated horse power is certainly the most convenient and probably the most practical method of stating the amount of work that is done in the engine; but *for gas and oil engines, it is possible to use another method of stating the power of the engine; a method which is not only more convenient and practical but which also gives more information as to the real actions taking place, and forms a better basis for the comparison of the thermodynamic performance of engines of different types.*

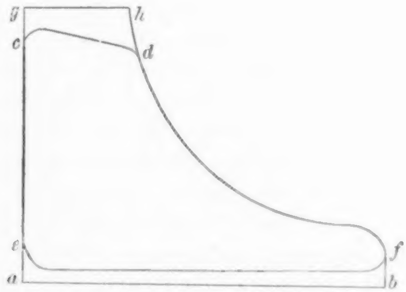


FIG. 1 INDICATOR CARD OF STEAM ENGINE

5 The indicated horse power of a steam engine is really the algebraic sum of two quantities; these are (*a*) the total work done by the steam *inside the cylinder* and (*b*) the negative work done in overcoming the frictional and inertia resistances of the steam during the exhaust period. Thus in Fig. 1 the total work done by the steam *in the cylinder* is *a c d f b*, and of this the amount *f e a b* is used up in overcoming the resistances to the escape of the steam, leaving the area *e c d f* as the indicated work, or the total work done on the piston.

6 The work done in the cylinder is not however the total work

done by the steam, since the steam has to do work in order to overcome the resistances to its admission. In Fig. 1, if *g h* represents the boiler pressure, the area *ghdc* is the work that the steam has to do in order to flow from the boiler to the cylinder. The total work done by the steam is consequently the area *g h b a*; and the area *c d f e* (i. e., the indicated work) is the difference between the total work done by the steam and the work necessary to get the steam into and out of the cylinder. The total work done by the steam in a steam engine is not shown directly on the indicator card, but has to be obtained by

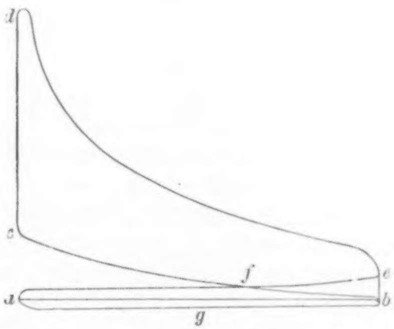


FIG. 2 INDICATOR CARD OF FOUR-CYCLE GAS ENGINE

drawing in the boiler pressure line and prolonging the indicator card to meet it.

7 The indicated horse power of gas and oil engines is defined by the Code of 1902 of the Committee on Standardizing Engine Tests, as the power developed in the engine cylinder (the algebraic sum of positive and negative works) minus the power indicated in the separate compression or feed cylinders, if there are any.

8 In a four-cycle engine, according to this definition, the indicated work is equal to the difference between the areas *c d e f* and *f a g b*, Fig. 2. If the area *e f b* be added to each of these, the indicated work is seen to be the difference between the areas *c d e b* and *e a g b*. The area *c d e b* is the total work done by gas; the area *a g b* is the work done in overcoming the resistances to the admission of gas, and the area *e a b* is the work done in overcoming the resistances to the exhaust of the gas; or in other words the area *e a g b* represents the work that has to be done to get the charge into and out of the cylinder. The indicated work of a four-cycle engine is consequently

seen to be the difference between (1) the total work done by the gas, and (2) the work necessary to get the gas into and out of the cylinder.

9 In a two-cycle engine with separate air and gas pumps, or with preliminary compression of the charge in the crank case, the indicated

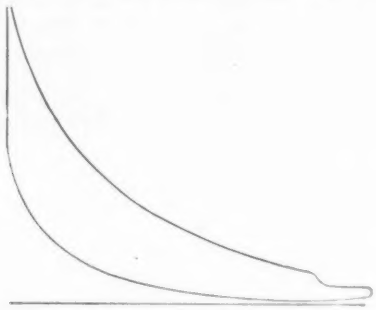


Fig. 3 Indicator Card of Two-cycle Gas Engine

horse power, according to the definition, is the difference between (1) the main cylinder horse power, Fig. 3, and (2) the indicated horse powers of the air and gas pumps, Fig. 4 and 5, or of the crank case, Fig. 6. In

Fig. 4 Indicator Card of Air Pump

this engine the exhaust occurs only near the end of the stroke, so that the amount of work done by the main cylinder piston in overcoming the resistance to the escape of the gases is so small as to be practically

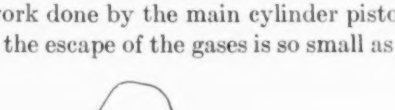


Fig. 5 Indicator Card of Gas Pump

negligible. The work represented by Fig. 3 is the total work done by the gas; while the work represented by Fig. 4, 5 and 6 is the work done in overcoming the resistance to the admission of the charge and consequently, in part, the work done in overcoming the resistance

to the exhaust, since the incoming charge helps to force out the exhaust gases. The indicated horse power of a two-cycle engine is seen to have practically the same meaning as the indicated horse power of a four-cycle engine.

10 In a Diesel motor, the conditions are the same as in an ordinary four-cycle engine, with the addition that work is done in compressing the air used to spray the fuel. The indicated horse power of the air compressor must, according to the definition, be subtracted from the indicated horse power of the main cylinder in order to obtain the indicated horse power of the engine. The difference between the areas *c d e b* and *e a g b*, Fig. 7, is the indicated work of the main cylinder, and, as with the four-cycle engine, Fig. 2, it is the difference between the work done by the gas in the cylinder and the negative work

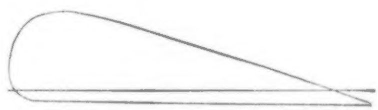


Fig. 6 Indicator Card of Crank Case

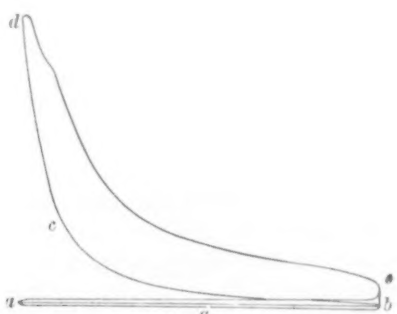


Fig. 7 Indicator Card of Diesel Engine

done in overcoming the admission and exhaust resistances. The compressor card, however, Fig. 8, is different from the compressor cards for the two-cycle engine, Fig. 4 and 5, for it represents not only the work required to overcome the frictional resistances to admission of the fuel spray to the cylinder, but also the work of compressing the air used for spraying, up to the pressure existing in the cylinder during the admission of the charge. It is obvious that if a large percent-

age of the air used per cycle in a Diesel motor were compressed in the air compressor instead of in the main cylinder, there would be a serious error in regarding the work done in the compressor as part of the frictional resistance to the admission of the fuel. In actual engines, the indicated work of the compressor pump is generally at least 6 per cent of the indicated work of the main cylinder. It is easily possible by drawing the cylinder admission pressure line *c d* on Fig. 8 to separate the work done there into its two components; the work done in compressing the charge (area *b*) and the work done in overcoming discharge resistances (area *a*). The frictional resistance to the admission of air to the compressor is too small to be shown on the diagram. The total work done by the charge is then the algebraic sum of the positive area *c d e b*, Fig. 7, and the negative area *b* Fig. 8, the work done in overcoming frictional resistances is the sum of the areas

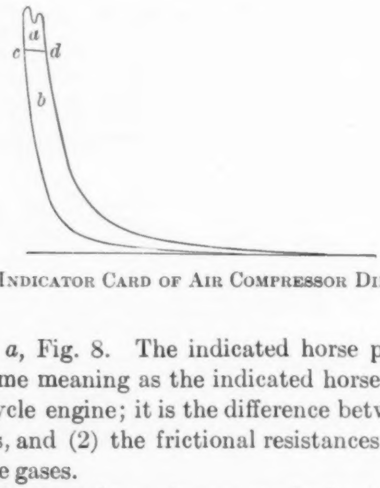


FIG. 8 INDICATOR CARD OF AIR COMPRESSOR DIESEL ENGINE

e a g b, Fig. 7, and *a*, Fig. 8. The indicated horse power of a Diesel motor has the same meaning as the indicated horse power of a four-cycle or a two-cycle engine; it is the difference between (1) the total work done by gas, and (2) the frictional resistances to the admission and exhaust of the gases.

11 In the analysis of the performance of a heat engine, there are two principal quantities that the engineer wants to know, namely, (*a*) the thermodynamic efficiency of the engine, or the percentage of the total heat going to the engine that is actually converted into work, and (*b*) the net efficiency of the engine, or the percentage of the total heat going to the engine that is available for doing useful work. The difference between these two efficiencies is the percentage of the total heat going to the engine that has been used up in overcoming the various resistances which the engine itself offers to the carrying out of the cycle of operations.

12 It has been the practice in the past to calculate the thermodynamic efficiency by finding the ratio of the indicated work to the total heat supplied. But this does not really measure the percentage of the total heat that has been converted into work; it measures the percentage of the total heat that is available for doing work after certain engine resistances, viz: those offered to the admission and exhaust of the working substances, have been overcome. The thermodynamic efficiency of an engine should have but one meaning and that is *the efficiency of the engine in converting heat into work*, irrespective of whether that work is used up, in part, in overcoming engine resistances or remains entirely available for useful applications. That is the plain meaning of the term and the only meaning which will permit a direct comparison of the efficiencies of the processes actually occurring in the cylinders of different engines If the indicated work is used in calculating the thermodynamic efficiencies, such a comparison does not necessarily throw any light on the actual processes at all, since the frictional resistances resulting from a poor design of compression pumps, valves, ports, etc., may more than offset the gain from the use of a more efficient cycle.

13 It is, moreover, important that the thermodynamic efficiency should have the suggested meaning in order to permit a fair comparison with the ideal cycle To state that the thermodynamic efficiency of a gas engine is 60 per cent of the thermodynamic efficiency of the ideal gas engine working under the same external conditions, is entirely misleading, if the gas friction resistances to admission and exhaust have been subtracted from the total work done by the gas. In some engines, the gas friction resistances may amount to 15 or 20 per cent of the total work, and if that were the case, an actual ratio of efficiencies of 70 per cent would appear to be but 60 per cent, that is, the apparent possibility of improvement of the purely thermodynamic processes would be reduced from 40 per cent to 30 per cent. If the gas friction is taken into account in calculating thermodynamic efficiencies, there does not seem any sufficient reason why the machine friction should not similarly be taken into account. The process of getting the charge into and out of the cylinder is purely mechanical—it is not part of the thermodynamic cycle.

14 The writer believes that for gas and oil engines, the power of the engine can be most usefully expressed in terms of total work done by the working substance—this might be called the *total horse power*, or, since it measures the amount of heat converted into work, the *thermodynamic horse power*. The total work for a four-cycle engine is the

area *cdeb*, Fig. 2; for a two-cycle engine, the area of Fig. 3; and for a Diesel engine, the area *cdeb*, Fig. 7, minus the work represented by the area *b*, Fig. 8, of the air compressor card. As measured in this manner, the total work is not entirely independent of the design of exhaust valves and passages, since the occurrence of release before the end of the stroke (which is necessitated by the resistances of the exhaust) reduces the total work area. It is only in the case of the comparatively early exhaust of the two-cycle engine that the actual work might be considered as being affected in an appreciable manner by the release before the end of the stroke. It is, however, proper to regard the work of this cycle as being finished when the exhaust opens—the toe of the diagram being the equivalent of the negative area of the four-cycle diagram. Since the area of the toe of the diagram is always extremely small, its inclusion in the total work area introduces no appreciable error.

15 The total work done by the working substance is used up in three ways:

a In overcoming the resistances to the admission and exhaust of the charge; this may be called *gas friction work.*

b In overcoming engine friction (this may be called *machine friction work*).

c In doing *useful work.*

16 The indicated horse power is then the total horse power minus the gas friction horse power and it retains the meaning it has always had.

17 An ordinary gas engine test permits the determination of the total horse power, the gas friction horse power, the machine friction horse power and the useful or brake horse power. The value of finding these separate horse powers will be apparent if, for example, a comparison is to be made between two-cycle and four-cycle gas engines. It is urged against the two-cycle engine that it obtains its very great advantage of nearly doubled power per cubic foot of piston displacement, at a cost of considerable loss in efficiency. This loss in efficiency is said to be (1) thermodynamic, resulting from (*a*) the loss of some of the charge to the exhaust during admission, or (*b*) the retaining of too much of the burnt gases in the cylinders; (2) gas friction loss resulting from the separate compression of the gas and air and the consequent extra valve and pipe resistance; and (3) machine friction losses resulting from the actual mechanical arrangements. The statement of the separate horse powers will throw light at once upon all

these points, and will show also wherein any particular engine fails to come up to the standard of its class.

18 From the commercial point of view, there is no advantage in retaining the indicated horse power, since it is the brake horse power that the engine user wants. From the scientific point of view, the indicated horse power can be of use only for the comparative study of engines and if it is not the best measure of power for that purpose, it should not be permitted to retain its present position.

19 If the total horse power, gas friction horse power, machine friction horse power and brake horse power are used as the standard measures of the engine power and losses, the various engine efficiencies could be defined in the following manner:

$$\frac{\text{Total work}}{\text{Work equivalent of total heat supply}} = \text{thermodynamic efficiency}$$

$$\frac{\text{Brake work}}{\text{Total work}} = \text{engine efficiency}$$

$$\frac{\text{Brake work}}{\text{Total work—Gas friction work}} = \frac{\text{Brake work}}{\text{Indicated work}} = \begin{matrix}\text{Mechanical}\\ \text{efficiency}\end{matrix}$$

$$\frac{\text{Brake work}}{\text{Work equivalent of total heat supply}} = \text{Net efficiency}$$

$$\text{Thermodynamic efficiency} \times \text{engine efficiency} = \text{Net efficiency}$$

20 These definitions retain for indicated horse power and mechanical efficiency their usual meanings.

21 The thermodynamic efficiency is the actual efficiency of the process of converting heat into work; the engine efficiency is the true measure of all the frictional losses of the actual mechanism, not only the friction of bearings and pistons, but also of the gas entering and leaving the cylinder; and the net efficiency is the quantity that interests the person who pays the bills for fuel. The mechanical efficiency has its use in showing the extent of machine friction losses, but unless the engine efficiency is also stated, it tends to obscure the real magnitude of the more or less avoidable friction losses in an engine.

22 There would be certain incidental advantages from the use of total horse power as the unit of measurement apart from the more important scientific advantages of a unit which means a single definite thing—and not the sum of two quantities of very different kinds. In

ordinary practice, there is more complexity and greater possibility of inaccuracy in the measurement of the indicated horse power of gas and oil engines than is the case with steam engines. The greater complexity arises from the fact that it is necessary in the two-cycle engine to take indicator cards not only from the main cylinder but also from the auxiliary gas and air pumps or from the crank case, and for a Diesel engine, it is necessary to take cards from the air compressor as well as the main cylinder. The greater inaccuracy results from the fact that in going around the *negative* area of the four-cycle, or Diesel-cycle cards, the probable planimeter error has the same absolute magnitude as in going around the *positive* area, and these two errors may both be of the same sign. If, to avoid this, a weak spring diagram is taken of the work of the exhaust and suction strokes, we have the complexity of another indicator. Of course when scientific results are needed, in which case the gas friction horse power must be obtained, it will be necessary to take cards from all the auxiliary cylinders, and the greater complexity cannot be avoided; but for ordinary commercial purposes, if any measurement of power is required beside the brake horse power, the total horse power would serve quite as well as the indicated horse power, and it could be obtained more easily and with more accuracy. Commercially, the indicated horse power is of no particular use when the brake horse power is known, and scientifically it is less useful than the total horse power.

23 The proposed new measure of power cannot be applied conveniently to the steam engine, nor does it seem desirable so to apply it, since the practice in that case is firmly fixed. In the steam engine, part of the compression work is carried out in the air and feed pumps, but the indicated work in these auxiliaries is not taken into account in calculating the indicated horse power: i. e., a different practice exists from that which the Society recommends as proper for the determination of the indicated horse power for gas and oil engines. The history of the steam engine probably lies more in the past than in the future, so that a change in practice is not particularly desirable, even if practicable: but history of the gas and oil engines almost entirely in the future, and a proper choice of the units of power may help to make that history more clear.

24 In conclusion, the writer wishes to submit to the Society the desirability of an early revision of the code of rules for carrying out and reporting gas and oil engine tests. The remarkable extension in the use of the gas engine, the growth of the large variety of types

which it has stimulated, the considerable body of research throwing light on that form of motor which has been published since the appearance of the code, have made it apparent that the code is deficient in certain respects, and have rendered many changes desirable.

25 When such revision is made, the writer hopes that there may be incorporated in it the suggestions as to horse power and efficiencies which he has presented above.

DISCUSSION

Mr. Henry Harrison Suplee As a discussion upon the paper of Prof. Lionel S. Marks, the secretary of the Gas Power Section of the Society was requested to contribute to the published discussion an abstract of the controversy which took place before the *Verein deutscher Ingenieure*, concerning the true mechanical efficiency of the modern internal combustion engine.

2 As that discussion covered many pages of the *Zeitschrift* of that Society, and aroused a somewhat acrimonious controversy, it has been thought best to give simply a summary of the essential points involved, as forming an introduction to the paper of Professor Marks, and to refer those who may be interested in the details of the discussion to the reports of the German Society.

3 At the forty-fifth annual convention of the *Verein deutscher Ingenieure*, held at Frankfort-am-Main, on June 6, 1904, Professor Riedler delivered an address upon the subject of large gas engines, in the course of which he called attention to the fact that the determination of the mechanical efficiency of such machines presented important differences from the accepted conditions for the steam engine.

4 In the case of the steam engine, the mechanical efficiency is taken as the ratio between the brake horse power and the indicated horse power. For the earlier four-cycle gas engines this practice had been followed without comment. When, however, the two-cycle gas engine came into use, the charge of air and gas was partially compressed in a separate cylinder, and the question arose as to whether the work absorbed by this compression cylinder should be deducted from the indicated power of the engine, or whether it should be left as a separate matter. Professor Riedler cited a report made by Professor Meyer, of Berlin, upon a 500-h.p. Oechelhauser engine, in which the pump horse power had been subtracted from

the indicated horse power before the computation of the mechanical efficiency.

5 That this question possessed more than an academic significance appears in the fact that had the resistance of the compression pumps not been deducted from the indicated power before the computation of the mechanical efficiency, the latter would have been nearly 8 per cent lower than appeared in the report of Professor Meyer.

6 The discussion upon the paper of Professor Riedler led to the appointment of a special committee, consisting of Messrs. Schöttler, Stodola and Schröter, to determine the correct procedure in such cases. This committee reported in favor of the procedure of Professor Meyer, while at the same time rather evading the question as to the scientific basis for his method.

7 Professor Riedler returned to the charge, and maintained that the indicated power is the actual power developed in the cylinder and that the work absorbed by the pumps constitutes a part of the resistance of the engine.

8 Professor Stodola, whose views are certainly entitled to full consideration, maintained that Professor Meyer's practice conformed to that obtaining with the steam engine, at least so far as the high-pressure engine is concerned. With the condensing steam engine there is a difference in practice: if the air pump is driven by the engine its resistance is not deducted from the indicated cylinder power; if the air pump is independently driven, its resistance is subtracted. In general, Professor Stodola was inclined to believe that, for two-cycle gas engines, the pump resistance should be deducted; the mechanical resistance of the engine being assumed to include only frictional resistances, so far as the computation of the mechanical efficiency is concerned.

9 Mr. Diesel suggested the subdivision of the various quantities, thus: indicated power = the full power represented by the indicator diagram of the working cylinder; useful power = the power delivered at the engine shaft; effective power = the sum of the useful power and the pump resistance; mechanical efficiency = useful power divided by indicated power (thus supporting Professor Meyer); dynamic efficiency = effective power divided by indicated power; pump factor = pump resistance divided by the indicated power.

10 The whole subject resolved itself into a matter of definition, as to whether the compression pumps formed an integral part of the engine or whether they should legitimately be considered as auxiliaries.

11 For a full report of the matter reference should be made to the *Zeitschrift des Vereins deutscher Ingenieure* for 1905, as follows:

a Original paper of Professor Riedler; Report of Professor Meyer to Messrs. A. Borsig upon a 500-h.p. Oechelhauser gas engine; Report of Messrs. Schöttler, Schröter, and Stodola;

b Reply of Professor Riedler; all in the issue of February 25, 1905;

c General symposium upon the mechanical efficiency and indicated power of the gas engine, including contributions from Messrs. Stodola, Riedler, Schöttler, Meyer, Ehrhardt and Wagner; issue of April 1, 1905;

d Communication from Mr. Rudolph Diesel upon the mechanical efficiency and indicated power of the gas engine; issue of May 20, 1905;

e Communication from Prof. Hugo Guldner; issue of June 24, 1905.

The Author The brief abstract of the German discussion on the mechanical efficiency of gas engines, supplied by Mr. Suplee, shows the confusion which exists at present in the statement of the power measurements and efficiencies of these motors. The vote at the Detroit meeting for the appointment of a committee to formulate a code of rules for testing gas engines gives the Society the opportunity to prevent that confusion in American practice. The suggestions which I have made, if accepted, would establish a practice different from the current practices of both Germany and England. It would make the statements of gas engine performances perfectly definite, practically and scientifically useful, and immediately available for comparison with other engines.

No. 1203

A SIMPLE METHOD OF CLEANING GAS CONDUITS

By W. D. Mount, Saltville, Va.
Member of the Society

Three years ago the writer had occasion to ask for proposals on a battery of gas producers, including the design of the distributing conduit and branches, though the latter were to be built under separate contract, and by parties other than the contractors for the producers.

2 As the producers were to supply gas to furnaces in continuous operation over long periods, which, for efficient working, required very uniform conditions of heating, the question of an uninterrupted flow of gas of uniform quality became of exceptional importance. We found that all of the builders of producers whom we asked to submit specifications and quotations could, in a general way, meet the conditions of uninterrupted service; that is, it would be necessary to shut the gas off a few hours once each week, and clean the accumulated soot out of the conduits.

3 Our idea of continuous service was somewhat different. It did not mean shutting down once a week, or once each month, but meant a condition of operation absolutely without interruption for an indefinite period, and we finally made this a condition of our acceptance of the contract.

4 The installation was to be in a department already in operation for several years, and where the use of gas was not originally contemplated; therefore, the distributing conduits had to be designed to meet existing conditions of space and apparatus in a department already much overcrowded, so much so in fact that a duplicate system of conduits, one to be in use while the other was being cleaned, designs for which were submitted by one builder as the only possible solution of the condition of uninterrupted service, was entirely out of the

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

question, to say nothing of the greatly increased cost of installation.

5 In placing the contract our decision was not based altogether on the merits of the different producers offered, but largely on the method or system of cleaning, and a design of distributing conduits which seemed to provide the greatest facilities for meeting the imposed condition of uninterrupted service.

6 After carefully canvassing the designs submitted, a contract was placed with the Morgan Construction Company, of Worcester, Mass., through their engineer, Mr. E. A. W. Jefferies, a member of the Society who, to quote from one of his letters, was "confident a method of cleaning the conduits, which, because of conditions imposed, was the most important function to be provided for adequately, could be worked out, but it required special study, and involved some features not heretofore fully developed." I may say at this point that the congested condition of the department in which the producer plant was to be installed, as well as the location of existing apparatus, precluded the possibility of using underground conduits, and made it necessary that an overhead system be adopted, and, as already stated, the lack of room prevented the system from being in duplicate.

7 The plan of action arranged, as outlined by Mr. Jefferies, consisted in providing facilities in the way of openings in the conduit for blowing depositions of soot through into the stack, by means of steam jets, connections to the base of the stack being made from each distributing conduit.

8 The above outlined operation was not to occupy more than fifteen minutes, during which time the gas was to be shut off, but the heat in the furnace was not to be seriously impaired. The condition of uninterrupted service, it is to be noted, was not fully met, and the writer was confident at the time, and subsequent developments proved his prediction to be correct, that when the gas was shut off, even for as short a time as 15 minutes, it meant shutting off the product from the furnaces, or in other words, it meant an absolute shutdown of the department during the operation of cleaning.

9 The producers and distributing conduits, however, were installed as designed, and the proposed method of cleaning put into effect, with the exception that compressed air was used in place of steam. Measured by experience gained, the method was a success, but as a means of keeping the gas conduits clean, it was a failure.

10 It did not take long, however, to determine that the idea was all right, and that our lack of success was due to the fact that the

facilities provided in the design for applying the air or steam were inadequate, as well as improperly located.

11 The experience gained in the early days led finally to the development of the system to its present form, which is fully detailed in the accompanying sketches, Fig. 1 and Fig. 3, and which, I am glad to report, has been so successful that for the past 18 months we have

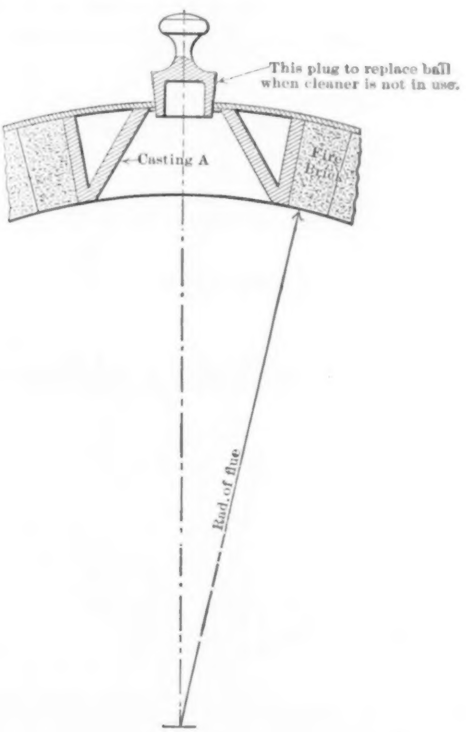


FIG. 1 CLEANING HOLE CASTING

never for a moment had the gas shut off from our distributing conduits for the purpose of cleaning out the depositions of soot.

12 The method and apparatus, as will be noted from the drawings, are extremely simple. We have so much confidence in compressed air that we are almost inclined to believe that it is essential to the successful operation of the method. The air should be thoroughly

dry and at not less than 80 pounds pressure. We find in practice that the connections from the conduit to the base of the stack are, for cleaning purposes at least, entirely unnecessary. The only time they are used is in making repairs on the conduits when they are opened for the purpose of drafting out the gas. We also find that about 75 per cent of the soot is deposited in the drop legs immediately behind the producers, the balance being dislodged by means of the air jet from the brick lining and carried along with the current of gas to the furnaces, where it is almost wholly consumed.

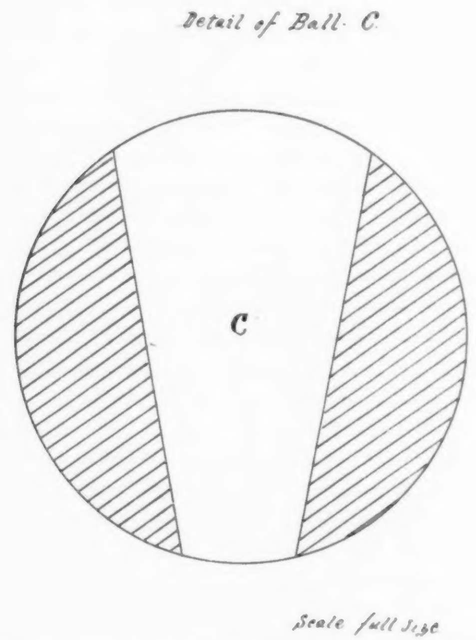


FIG. 2 DETAIL OF FIG. 3; BALL JOINT FOR AIR PIPE

In Fig. 3 A A are openings in specially designed castings shown in detail in Fig. 2, spaced along the top of the conduit and of the same depth as the brick lining. C is a cast iron ball of a diameter sufficient to close the opening in A and free to slide on the one-half inch bent pipe B, Fig. 3, which is connected by hose to the air main; a one inch pipe serving also as one of the hand rails along the top of the conduit.

14 The openings *A A* are spaced at a distance nearly equal to the diameter of the conduit (although this spacing is not necessarily fixed), which we have found by experience to be about right for effective work; in any event they should not be at a distance exceeding six feet.

15 The bent pipe *B* should be long enough to reach all parts of the conduit from *A* to *A*. The pipe, however for convenience in handling cannot be over eight feet and its length, therefore, in a measure, fixes the distance from *A* to *A*. Permanently attached to the end of the pipe is a short length of air hose, having attached at the other end the male portion of a standard one-half inch Joy coupling. Located along the hand rail air pipe, at distances from two to

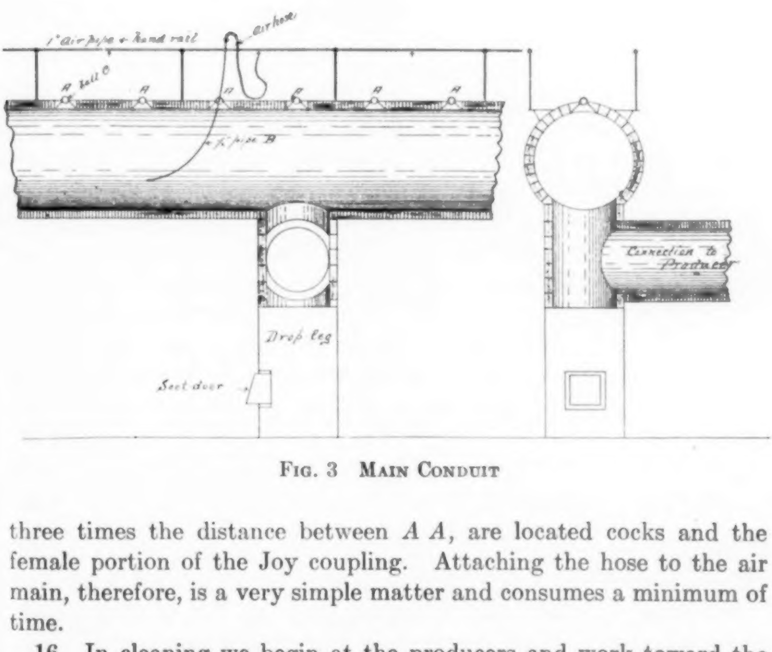


FIG. 3 MAIN CONDUIT

three times the distance between *A A*, are located cocks and the female portion of the Joy coupling. Attaching the hose to the air main, therefore, is a very simple matter and consumes a minimum of time.

16 In cleaning we begin at the producers and work toward the furnaces. To the casual observer it would seem that the operator did nothing more than stick the bent pipe through the opening *A* and turn on the air. This, however, is, of course, the most insignificant part of the operation. It is necessary that the pipe be given a circular motion, which will bring its discharge end in contact with all parts of the brick lining from one opening to another. A careful, conscien-

tious workman will, in a little time, become very expert in the operation, and when the work is properly done it can be depended upon to remove all depositions of soot.

17 In conclusion, it is only necessary to say that the gas is not shut out of the conduit during the operation of cleaning, nor is its flow in any way checked. The producers do not seem to be affected in the slightest degree; in fact cleaning is conducted absolutely without interruption to producers, conduits and furnaces, and that it is efficient is evidenced by the fact that when our conduits have been opened for repairs we have always found them clean to the brick lining.

No. 1204

THE SURGE TANK IN WATER POWER PLANTS

A DEVICE FOR AID IN SPEED REGULATION AND PRESSURE RELIEF IN WATER POWERS WITH LONG PRESSURE PIPES AND HIGH VELOCITIES

By Raymond D. Johnson, Niagara Falls, N. Y.

Non-Member

INTRODUCTION

I have prepared this paper at the request of Mr. John R. Freeman and as an outcome of certain practical problems which I have assisted him to treat mathematically and which arose successively in planning three large hydro-electric developments. Therefore, although some may regard the paper as overflowing with equations and calculus, I trust it will nevertheless be regarded as practical.

2 "Surge Tank" is a term applied to a stand pipe or storage reservoir placed at the down-stream end of a closed aqueduct to prevent undue rise of pressure in case of a sudden diminution of draft, and to furnish water quickly when the gates are opened, without having to wait for the velocity in the long feeder to pick up. When such a device terminates a pipe used to feed water wheels, the changes of load, producing corresponding variations in gate opening, cause the stored water to rise and fall in wave-like swells or "surges;" hence the name. It has long been recognized by hydraulic men that a storage pond or artificial lake at the end of an aqueduct serves to remove harmful results of sudden changes in draft which would otherwise give rise to undue pressures, known variously under the names of "water-hammer," "breathing," "surging," etc., according to the nature and intensity of the pressure waves. So far as I know however, no attempt has previously been made to proportion such a reservoir economically and accurately. This I have tried to do.

3 I have also gone a step further in pointing out the efficacy of

Presented at the Detroit Meeting (June 1908) of The American Society of Mechanical Engineers.

the differential principle, whereby the area of the reservoir is lessened by 50 per cent and the regulation is markedly better. This principle is named from its action in differentiating the "pressure water" from the "stored water," which are identical in the ordinary tank. That is, the level in the tank is *both* the accelerating level, which urges the water to flow faster or slower in the conduit, *and* the level of the "stored water" as well. It is clear that the velocity in the conduit cannot be altered unless and until the level changes in the tank, and that the rapidity of such velocity change is proportional to the amount of such level-change. It therefore follows that the capacity would have to be much greater than it would if the accelerating head could instantly be applied to the pipe, without having to wait for the water level to be drawn down.

4 This desirable result can be had by drawing the water from the tank through constricted openings of fixed size, properly proportioned. A forced loss of head is thus effected, causing a sudden change in the accelerating level to a value which holds constant during the period of velocity change and until the level in the tank becomes coincident with the level represented by the accelerating head, which coincidence obtains simultaneously with the acquirement of the new velocity value as demanded.

5 The importance of a proper study of this subject was from time to time brought forcibly to the attention of Mr. Freeman when inspecting certain of the recent large water power plants in the West and in Mexico, where, in a dry country, water was almost continually wasted in order to regulate the speed of the water wheels, under their varying loads. In certain of his own designs he departed from the ordinary methods, in order to prevent waste of water and all needless waste of fall, and planned to use the water at high velocity through long pressure tunnels under the full mill-pond pressure. He provided for a surge shaft and surge reservoir close to the power house for the relief of the pressure due to arrest of motion of the great mass in the moving water column or for a momentary supply in case of increased demands while the elastic wave was transmitting the demand for more water back to the conduit entrance and overcoming the inertia of the water column.

6 It was easy to see that with a large surge reservoir near the power house this method would be simple and effective; but a large surge reservoir would be very expensive, and it appeared somewhat difficult to tell just how small this could be made without the sacrifice of quality of speed control. Also, on a canyon-side or in

other situations where natural support for a great weight at the proper elevation is lacking, the problem of minimum size of the surge tank becomes of great importance.

7 It may be stated at the outset that it is seldom or never necessary to waste any water in speed regulation, and it is usually very undesirable to do so. Governor-operated by-pass valves and deflecting nozzles have seldom an excuse for existence except, perhaps, in instances where a choking of the turbine feeder would cause the water to overflow the sides of an open flume and waste in any event; but with closed pipes feeding the wheels, however long, there need never be any substantial waste of water for regulation or pressure relief. It is, moreover, exceedingly cheap to effect this saving.

8 It is always more convenient if there is high-lying ground near the power house on which to locate the surge tank so that it may be operated under atmospheric pressure; but usually the dimensions of the tank are so small that it is feasible to locate it at any convenient lower level, even directly adjacent to the power house if necessary, by simply increasing the air pressure by closing in the top of the chamber and connecting it to a compressed air tank of adequate though comparatively small dimensions.

9 Where there is high lying ground somewhat remote from the power house, an atmospheric regulator may there be installed, and then, if the length of closed pipe between this point and the power house is too great to permit good regulation, a small secondary auxiliary compressed-air regulator may be placed adjacent to the power house. These two will work together perfectly, each attending to its own length of conduit.

10 I will not describe at this juncture the several possible forms and details of surge tank regulators, but, in order to make clear the conclusions which I have reached, will simply express forcibly my belief that any form of turbine-governing device which uses as its chief means of speed control the waste of valuable water, is, in general, a bad design and so far as I can at present see, ordinarily unwarranted and unnecessary.

CONCERNING CERTAIN POPULAR MISCONCEPTIONS

11 If excuse is needed for presenting this paper, it can be found in the published statements of certain experienced hydraulic engineers, and in the design of certain otherwise admirable plants.

12 For example, the writer has recently noticed, in Mr. H. E. Warren's very able article written for the Lombard Governor Co.

only a year ago, a statement in reference to the effect of long pressure conduits upon regulation, that "The only complete remedy for the troubles in speed regulation caused by excessive inertia of a water column is some form of by-pass valve directly connected with the water-wheel gates, arranged to open as the gates close, and thus keep the velocity of the column nearly constant." When such a statement is published by people who make water-wheel regulation a business and have a reputation for success, it is well to demonstrate the possibilities of an opposite course.

13 The remedy of a by-pass as suggested is wasteful of valuable water which would otherwise be storing up in the pond. It will be shown that speed regulation is possible without a by-pass and its waste. The article cited further states that "Although it is possible to prevent excessive rise of pressure due to reduction in velocity of long water columns, no means have been devised to produce quick acceleration of the water when, because of a sudden increase in load, the speed of the turbine begins to fall." The writer believes it can be demonstrated that this statement is not altogether correct, and will describe a form of differential regulator such as to provide acceleration "quick" enough so that by the time the automatically stored water is used up, the velocity in the long pipe will have acquired its new demanded value. This paper is intended to elucidate the mathematical theory of such action. Since the theory of this device was worked out some six years ago, the writer perhaps ought to apologize for not presenting it sooner to his fellow designers for their criticism and suggestions.

14 The problems of water turbine governing[1] will not be treated in this paper, as will later become apparent, but the argument will be confined to showing how simple it is to regulate automatically the acceleration and retardation in a long column of water so that the difficulties in turbine governing largely disappear.

15 It is never required of a large plant to furnish a sudden increment from zero to full load, or anything like it, although with a proper surge tank this could be easily done with no more than 10 per cent variation in head in a 5000-foot feeder at 8 ft. velocity, if desired, instead of a variation of some 160 per cent stated as unavoidable in the paper cited. As a matter of fact, all that would be necessary, if the topography of the country was such that an elevation equal to that of the pond water could be had within a reasonable distance of

[1] Thorough and presumably correct methods for working out such problems have been developed by the writer, but they are of minor importance when the water column is properly regulated.

the power house, would be to install a small tank, of less than three times the diameter of the feeder pipe. For example, in the case cited in the paper referred to, the pipe was 5 ft. in diameter, in which case the regulating surge tank need be no more than 15 ft. in diameter, and the cost of this is obviously a small matter if it is successful in reducing changes in head from an imaginary 160 per cent down to 10 per cent of the assumed head of 400 ft. Obviously 160 per cent of the head cannot be attained in acceleration.

FUNCTION OF A STANDPIPE

16 Where the topography of the country is not such that a high elevation can be had near the power house, compressed air must be resorted to as will be later pointed out.

17 Let us briefly review just what the function of a stand pipe is, and how it operates to supply water to the wheels while the long conduit is changing its velocity from a lower to a higher value as demanded by a change in load at the power house.

18 To be most effective, the standpipe or surge pipe should be located as near the power house as possible. It is usually placed at the end of the conduit or main feeder, and is sometimes formed by simply turning up the end of the conduit, into a vertical or inclined position, and extending it up to a level somewhat above that of high water in the forebay or storage pond.

19 The penstocks or individual water wheel feeders are led out of the main feeder at points as near as possible to this standpipe. Where steel penstocks or wood stave pipe bound with iron hoops are used, plants are laid out, where possible, so that the greater part of the total drop occurs between the end of the main conduit and the power house, thus keeping the pressures in the main conduit as small as possible and permitting the emergency gates at the heads of the individual turbine feeders to work under moderate pressure. Care is also taken to have these separate turbine feeders as short as practicable.

20 Now, with this arrangement, when the water wheels are running under a constant load, it is clear that the water level in the standpipe will be stationary at an elevation below pond level equal to the friction head in the conduit, and this level will represent the effective head on the wheels, if the friction losses in the turbine feeders be for the moment disregarded.

21 Let us study in a casual way the cycle of phenomena following a sudden demand for more water by the wheels:

(*a*) We have assumed a conduit so long that acceleration in it is far too slow to meet the sudden demand for more water within any reasonable time, and therefore, the additional water required nearly all flows momentarily from the standpipe, thus depressing its level.

(*b*) This depressed level being below the end of the hydraulic gradient due to the existing velocity, a lessened pressure is transmitted back to the pond with the velocity of an elastic wave or the speed at which sound would travel in water, about 4500 ft. per second. This quickly produces an accelerating head which begins to increase the velocity of the water in the conduit.

(*c*) The water level in the standpipe continues to drop and the velocity in the conduit to increase until such increased velocity value becomes equal to the new demand for power. The larger the standpipe area, the less the level will have become depressed before this condition is reached, and therefore the less the variation in head that will be felt at the water wheels.

(*d*) The standpipe level, meanwhile, will have been drawn much below the elevation of the end of the existing velocity gradient, so that a greater acceleration than necessary in the conduit will be caused and must continue. The level in the standpipe therefore soon begins to rise and more water soon begins to be supplied than is being used by the turbines even under the increased load.

(*e*) The conduit velocity continues to increase until the water level in the standpipe has risen to an elevation identical with that of the end of the velocity gradient existing at that time, after which it begins to be retarded as the standpipe level continues to rise.

(*f*) The maximum elevation is reached when the surplus energy created in the conduit by the first depression of level is partially reconverted into potential energy represented by this temporary maximum elevation of water in the standpipe. Meantime, retardation of the conduit velocity continues, until

(*g*) The elevation of the water again becomes coincident with the end of the velocity gradient.

Neglecting for the moment the complications induced by the efforts of the turbine governor, this completes a description of one complete

cycle of the pressure wave, which goes on repeating itself indefinitely until smothered by the action of friction.

RESULT OF THE TURBINE GOVERNOR'S EFFORTS

22 Before going further we may take time to consider briefly in a general way the possible dangers of a standpipe of ordinary form and moderate size.

23 Where such a device terminates a long closed flume, it is clear that any demand for additional load in the power house will have to be fed primarily, and almost wholly, from the water contained in the standpipe. This creates a depression in the water level which in turn begins to accelerate the main column of water in the feeder. Now if there were no friction affecting this column there would be not only an ultimate recovery of the water level (which would be the equivalent of a pendulum returning to a vertical position), but there would be a subsequent surge of the water to a level as much above the original level as the depression was below it. If this surging were plotted with time it would give a perfect sine wave, which without friction loss, would never cease when once started by an increment of load. The effect of friction is to increase the throw of the first depression but to decrease in greater proportion the subsequent surge in such a way as to produce a damped harmonic which of course soon dies out, if the standpipe is large.

24 With a long conduit, a mile or more in length, these oscillations are comparatively slow, occupying sometimes more than one or two minutes for a complete oscillation, so that any ordinary water wheel governor has no difficulty in keeping step with the wave, gradually closing the gates as the head increases, and opening them, conversely, in an effort to keep the product of velocity, or cubic feet per second, and head nearly a constant quantity. This essential and unavoidable action of the governor may lead to astonishing results, on account of which the ordinary standpipe might prove almost worse than useless.

25 Returning to our first harmonic curve where friction was neglected, let us consider for a minute the effect of this governor action upon such a wave. It must be obvious that the further opening of the gates as the depression in level takes place will have the effect of increasing the amount of such depression. Now, without any closing of the gates, the surge will raise the level an equal distance above normal; but when we consider that the gates must close as the pressure rises, it is clear that the wave will rise a greater amount than

it fell, and so *go on in an ever-swelling harmonic*, ceasing to increase only when it reaches the top of the surge pipe and spills over. Then, after continuing to spill for some time, it will return in another dip only to rise again and spill some more, thus continuing to spill over at regularly recurring intervals so long as the load holds steady.

26 Thus far we have been neglecting friction in this last argument. A little thought at this juncture will lead one to a natural conclusion that if the swelling effect, due to the governor action, exceeds the damping effect, due to the action of friction, *the harmonic will swell in spite of the friction* and this is precisely what will take place *if the standpipe area is below a certain critical amount* which is mostly a function of the length of the conduit.

27 This unstable condition would make operation practically impossible, and it is an extremely easy thing to bring about such a condition simply by being a little careless in the choice of a regulating device. Furthermore, it does not necessarily follow that the standpipe area is sufficient simply because it is larger than this critical size. This is more likely to be true for high heads than for low heads, but often the variation in head due to throwing on or off part load may be too great a proportion of the normal head, even though an increasing wave cannot exist.

28 To sum up, therefore, it is extremely dangerous to design a stand pipe of moderate size without going thoroughly into all the conditions which it affects, unless the load is to be particularly steady and the main feeder pipe rather short. Where the feeder is more than a mile long the problem becomes tremendously important, and is worth careful study.

29 As we develop the formulæ for working out these problems, it will become apparent, as indeed common sense would indicate, that all the desired conditions may be fulfilled by simply making the standpipe of sufficient area, so the whole matter resolves itself into a question of area and depth.

30 It will later be shown what a great advantage can be gained over the simple standpipe or storage tank *by differentiating the pressure water from the stored water*. In practice, this may be easily accomplished by using a comparatively small standpipe or surge shaft which is contained within a larger tank, and communicating with it only by means of fixed ports or holes in the smaller pipe near the bottom of the larger one.

31 The small pipe may be at one side instead of within the tank and may communicate with it through a restricted by-pass of fixed size.

32 The action of such an arrangement is like that of a dash pot, and if properly proportioned it effectually damps any vibrations which would swell to undue proportions in an ordinary tank of twice the area, so that the cost of this style of combination regulator will usually be found not much more than one-half that of an ordinary simple storage tank of sufficient size to accomplish the same final result. Further, the regulation is much better, for there is little or no chance that bad conditions may result from throwing on loads at critical times so as to superimpose one wave upon another, as is quite possible with the simple tank.

THE MATHEMATICAL ANALYSIS OF THE PROBLEM

33 The best way to approach these problems is through equations of work and energy, and it may be as well to start out by laying down a perfectly general principle which will later become evident.

34 The work done within either the standpipe or differential regulator, in raising or lowering the water, is precisely equal to the work of lifting the weight of the water in the conduit through a distance equal to the head due to the velocity lost or gained in the conduit, which change in velocity is in turn due to a change in load. In what follows let

V_1 = initial steady velocity in the conduit before the increment of load is demanded (assumed constant in retardation also).

V_2 = new velocity due to load increment. This is assumed constant for convenience in calculation, and must be evaluated according to judgment. (See Par. 55.)

$V_2 - V_1$ = an ultimate change in velocity due to load change upon the assumption of constant draft velocity V_2 or V_1.

y = change in water level in tank during surge.

A = conduit area.

R = ratio of conduit area to standpipe area.

γ = weight of water.

L = effective length of conduit.

h = net working head for velocity V_1, neglecting penstock losses between surge tank and water wheels.

35 The work done within the standpipe, neglecting all friction, would be $\frac{1}{2} y_1^2 \times \frac{A}{R} \times \gamma$. The equivalent energy destroyed or gained in the conduit would be $\frac{(V_2 - V_1)^2}{2g} \times LA\gamma$, where $V_2 - V_1$ is the ultimate change in velocity due to the load change, for a constant value of V_2 and V_1.

36 Equating these we have

$$y^2 = \frac{R\,L}{g}\,(V_2 - V_1)^2 \text{ or } y = \sqrt{\frac{R\,L}{g}}\,(V_2 - V_1)$$

37 This formula neglects both friction and the bellows action of the wheel gates in keeping step with the wave. This latter consideration may sometimes be neglected if the regulator is *otherwise designed* correctly. (See Par. 55.) The equation given is not the equation of the wave curve, but simply an expression for y maximum. We will now develop the differential equation of this curve, still for the present neglecting friction, and also work out the period or time of vibration of the water pendulum.

38 The work done upon the water in the tank when part load is rejected is $y\,d\,y\,\frac{A}{R\gamma}$: this may also be written in terms of the velocity change or $\frac{\gamma LA}{g}\,(V - V_1)\,d\,v$, whence we have

$$y\,d\,y = \frac{L\,R}{g}\,(V - V_1)\,d\,v$$

$$\int_o^y y\,d\,y = \frac{L\,R}{g}\int_V^{V_2}(V - V_1)\,d\,v$$

Solving for y we have

$$y_r = \sqrt{\frac{R\,L}{g}}\left\{(V_2 - V_1)^2 - (V - V_1)^2\right\}^{\frac{1}{2}} \qquad [1]$$

Similarly for accelerating, when part load is demanded we have

$$y_a = \sqrt{\frac{R\,L}{g}}\left\{(V_2 - V_1)^2 - (V_2 - V)^2\right\}^{\frac{1}{2}} \qquad [2]$$

39 The maximum value of y can be seen to be the same in both cases, or as previously demonstrated,

$$y_1 = \sqrt{\frac{R\,L}{g}}\,(V_2 - V_1) \qquad [2a]$$

40 To find the expression for time in terms of velocity we have

$$\frac{L}{g} d v = y d t$$

$$\frac{L}{g} \int_{V_1}^{V} d v = \int_{o}^{t} y d t$$

or substituting for the value of y from [1]

$$\sqrt{\frac{L}{g R}} \int_{V_1}^{V} \frac{d v}{[(V_2 - V_1)^2 - (V_2 - V)^2]^{\frac{1}{2}}} = \int_{o}^{t} d t$$

from which

$$t_r = \sqrt{\frac{L}{g R}} \left\{ \frac{\pi}{2} - \sin^{-1} \frac{V - V_1}{V_2 - V_1} \right\} \qquad [3]$$

and similarly for accelerating by substitution from (2)

$$t_a = \sqrt{\frac{L}{g R}} \left\{ \frac{\pi}{2} - \sin^{-1} \frac{V_2 - V}{V_2 - V_1} \right\} \qquad [4]$$

We may now write the true eqûation of the wave curve by elimination of V between (1) and (3) or between (2) and (4) and we have, in either case the same equation:

$$t = \sqrt{\frac{L}{g R}} \left\{ \frac{\pi}{2} - \sin^{-1} \pm \frac{\sqrt{(V_2 - V_1)^2 - \frac{y^2 g}{R L}}}{V_2 - V_1} \right\} \qquad [5]$$

This is the relation between time and wave elevation and hence is the equation of the true harmonic curve. This expression is capable of ingenious simplification, as suggested by L. F. Harza, as follows:

$$y = Y \sin \frac{\pi}{T} t$$

where Y is y_{max} and $\frac{T}{2}$ is the time corresponding to Y. For y maximum we have simply,

$$t = \sqrt{\frac{L}{g R}} \left(\frac{\pi}{2} + \sin^{-1} 0 \right) \qquad [5a]$$

41 This is a periodic function, and denotes the successive times when the wave reaches its crest.

$$t \text{ for } y = 0 = \sqrt{\frac{L}{g R}} \left(\frac{\pi}{2} + \sin^{-1} 1 \right)$$

which is also a periodic function and denotes the successive times when the wave passes through zero. It is interesting to note that where $R = 1$ the formula is identical with that for a simple pendulum as might have been guessed. That is, the time of complete vibration of this water pendulum of length L is

$$\pi \sqrt{\frac{L}{g}}$$

where the stand pipe is the same size as the conduit. This time is precisely the same as for a pendulum of a length equal to that of the conduit, and is independent of the amplitude; that is, the amount of load thrown on or off cannot materially affect the period of surge.

COMPUTATIONS INVOLVING FRICTION

42 The introduction of friction into these computations leads to such difficult mathematics that it is almost necessary to resort to an approximation. This may be done in many ways which may be devised to order. I will set forth one of the most evident, which leads to no important error in any practical case and has the advantage of being comparatively simple.

43 Proceeding as before, the work done within the standpipe is no longer

$$\frac{1}{2} y^2 \times \frac{A}{R} \times \gamma$$

when we take friction into account, but is

$$\frac{A}{R} \gamma \int_o^y (y - Z)\, d\, y$$

where Z represents the change in hydraulic gradient and is equal to $C\ (V_2^2 - V^2)$ for retarding or $C\ (V^2 - V_1^2)$ for accelerating, C being the friction constant. The exponent 1.85 would probably be closer than 2 and may just as easily be used if desired. The expression is not as simple as before, for now we have

$$\int_o^y (y - Z)\, d\, y = \frac{L\, R}{g} \int_V^{V_2} (V - V_1)\, d\, v$$

$$= \int_o^y y\, d\, y - C \int_o^y (V_2^2 - V^2)\, d\, y$$

$$\int_o^y y\,d\,y = \frac{L\,R}{g}\int^{V_2}(V - V_1)\,d\,v + C\int_o^y (V_2^2 - V^2)d\;y$$

or

$$y^2 = \frac{R\,L}{g}\left\{(V_2 - V_1)^2 - (V - V_1)^2\right\} + 2\,C\int_o^y (V_2^2 - V^2)\,d\,y \quad [6]$$

44 Now $V^2\,d\,y$ in the last term is extremely difficult to integrate. But we know something about the value of $C\;(V_2 - V^2)$ in terms of y; not much to be sure, but this at least, that $y - C\;(V_2^2 - V^2)$, which is the accelerating head, is positive in the first cycle up to the time V reaches its maximum value, when the expression changes sign and remains negative for half a cycle, etc. We know, also, that y reaches its maximum some time before V reaches *its* maximum, and therefore that V reaches its maximum for a value of y between its maximum and 0. Also, the maximum of the accelerating head occurs somewhat before y attains its maximum. We may, therefore, write $C\;(V_2^2 - V^2) = X\,y$ where X is less than unity, up to the time V reaches its maximum, at which time it becomes unity. Now if we put $X = 1$ and $V = V_1$, we shall obtain the value of y maximum within a small amount.

45 A still closer approximation may be made, if desired, by putting $C(V_2^2 - V^2) = b\,y^n$, where b and n are constants. These would have to be evaluated by tedious graphical methods, and would eventually be found to modify the value of y maximum so little as to render the labor futile. Putting $V = V_1$ and $X = 1$ in equation (6) we have

$$y^2\text{ max.} = \frac{R\,L}{g}(V_2 - V_1)^2 + C^2\,(V_2^2 - V_1^2)^2 \quad [7]$$

which holds good both for accelerating and retarding. The time to reach y max. will be a little *longer* than that given by equations 4, 5, etc., but may be obtained with considerable accuracy by increasing the time in the same proportion that y increases when friction is considered. Dividing equation (7) by equation (2a) we have

$$1 + C^2\,(V_2 + V_1)^2\,\frac{g}{R\,L}$$

and multiplying this into the time of oscillation we have

$$t = \frac{\pi}{2}\sqrt{\frac{L}{g\,R}}\left\{1 + C^2\,(V_2 + V_1)^2\,\frac{g}{R\,L}\right\}^{\frac{1}{2}}$$

This may be written more conveniently

$$t = \frac{\pi}{2}\sqrt{\frac{L}{g\,R} + \frac{C^2\,(V_2 + V_1)^2}{R^2}} \qquad [8]$$

From this we see that t is still more or less independent of the amount of load change, but is decidedly affected by the actual values of V_2 and V_1. Therefore, we shall have different times of oscillation for the same actual velocity changes when starting out from different existing velocities. Practically, however, the value of the latter term is usually so small that the time of oscillation is really not a great deal affected by the friction, almost always under 10 per cent in an actual case.

46 This expression serves either in acceleration or retardation. The times in the two cases would probably be slightly different, and may be more nearly approximated in the following formulae, which are derived from equations (9) and (11) and are in terms of y max. as found from equation (7):

$$t_r = \frac{L}{g\sqrt{\frac{C\,y_1}{\sqrt{2}} - C^2\,V_2^2}}\left\{\tan^{-1}\frac{V_2}{\sqrt{\frac{y_1}{C\sqrt{2}} - V_2^2}} - \tan^{-1}\frac{V_1}{\sqrt{\frac{y_1}{C\sqrt{2}} - V_2^2}}\right\}$$

$$t_a = \frac{L}{2\,g\sqrt{\frac{C\,y_1}{\sqrt{2}} + C^2\,V_1^2}}\log_e\frac{(Z_1 - V_1)\,(Z_1 + V_2)}{(Z_1 + V_1)\,(Z_1 - V_2)}$$

where

$$Z_1 = \sqrt{\frac{y_1}{C\sqrt{2}} + V^2}$$

If reasonable numerical substitutions are made it will be found that t, t_r and t_a will be about the same in the three equations just given, seldom differing by as much as 5 per cent. The last two are based upon the fact that the introduction of a differential action due to the insertion of a surge pipe within a reservoir does not change the time of vibration, but only affects y, making it less in the proportion of 1 to $\sqrt{2}$, about. The friction itself does not make any enormous increase in the time, so that values found above will rarely exceed by 10 per cent those obtained from equation (5a).

THE DIFFERENTIAL REGULATOR

47 We now come to the development of the equations relating to the differential regulator. In one respect these are more readily determinable than the foregoing, inasmuch as the term containing the friction is easily integrated, but in order to get at the subject at all a preliminary assumption has to be made which is not quite true but which leads to insignificant and perfectly negligible errors.

48 This regulator, as previously described, consists of a small standpipe or riser communicating with a reservoir by means of fixed ports which compel a difference in level between the water surfaces in the two vessels when a change of load occurs. This initial separation of the two levels, due to the constriction at the ports, is so sudden as compared with the change of level in the *large reservoir* that it may be regarded as having existed previous to the load change, or as coming into existence simultaneously with the change of load, and having a value equal to the head required to force the excess or deficiency of water through the ports as designed. Furthermore, the level in the riser, once it has attained this value, will hold it throughout the accelerating or retarding period of the main conduit, *always provided* that there is just the right reservoir area so that by the time the reservoir level catches up to the riser level, the velocity in the conduit will have attained its new required value. Now, therefore, if we are going to wind up with the area accurately figured, we make no appreciable error in assuming the pressure level in the riser to be *constant* during the time the conduit velocity is picking up, and the excess water required by the wheels is being drawn through the ports from the reservoir.

49 That this condition of things actually does exist the writer has proved many times by a process of arithmetical integration; that is, computing the various curves representing changes in velocity, pressure head and friction head, by dividing the time into short spaces, say seconds, for these long swings, and laboriously computing, second by second, the successive values of acceleration, velocity, pressure level, friction head, water-wheel draught as the head varies, etc., thus integrating arithmetically all the various curves at the same time. This process is so obvious to anyone conversant with the ordinary physical laws, simply requiring a lot of patience, that it would be out of place here to go into a detailed description of the method. Let it suffice to say that *this method has one great advantage over calculus manipulation in that it makes possible the consideration*

of many more interdependent variables than could be handled by any ordinary man with his limited knowledge of calculus, and the writer is not really sure that the man lives who would not, if relying solely on the calculus, sooner or later have great difficulty in dealing with these multiplex phenomena. If the writer had not repeatedly proved his equations by just such tedious processes he would not now have the temerity to publish them.

50 Using the same nomenclature as before, and putting y max. $= y_1$, which as explained we are going to regard as constant, we have for retarding

$$d\,t = \frac{L}{g} d\,v \div [\, y_1 - C\,(V_2^2 - V^2)\,] \qquad [8a]$$

$$\int_o^t d\,t = \frac{L}{g}\int_{V_1}^{V_2} \frac{d\,v}{y_1 - C\,(V_2^2 - V^2)}$$

which is readily integrable and which gives an expression for the time it takes for the velocity to be retarded from V_2 to V_1. Integrating we have

$$t_r = \frac{L}{g\sqrt{Cy_1 - C^2 V_2^2}} \left\{ \tan^{-1} \frac{V_2}{\sqrt{\frac{y_1}{C} - V_2^2}} - \tan^{-1} \frac{V_1}{\sqrt{\frac{y_1}{C} - V_2^2}} \right\} \qquad [9]$$

Now if we multiply both sides of equation [8a] by RV we have

$$R\int_o^t V\,d\,t = \frac{R\,L}{g}\int_{V_1}^{V_2} \frac{V\,d\,v}{y_1 - C\,(V_2^2 - V^2)}$$

but

$$R\int_o^t V\,d\,t - R\,t\,V_1$$

is evidently the height which the water rises in the large reservoir, and since by hypothesis we are going to make the value of R such that this height will be equal to y_1 we may equate this expression to y_1, thus

$$y_1 = \frac{R\,L}{g}\int_{V_1}^{V_2} \frac{V\,d\,v}{y_1 - C\,(V_2^2 - V^2)} - R\,t_r\,V_1$$

which becomes after integrating and simplifying

$$y_1 = \frac{R\,L}{2gC} \log_e \frac{y_1}{y_1 - C\,(V_2^2 - V_1^2)} - R\,t_r\,V_1 \qquad [10]$$

in which t_r is known from equation [9]. This equation must be solved by successive approximations, but may be assisted very materially by remembering that

$$y_1 = \sqrt{\frac{L\,R}{2\,g}(V_2 - V_1)^2 + C^2\,(V_2^2 - V_1^2)^2} \text{ nearly,}$$

which may be directly solved, and the value of y_1 thus found may be substituted in (10) as a first trial.

51 The size of the ports will be determined later from a consideration of demanded loads rather than rejected loads. It is almost always true, however, that when the ports are properly designed for picking up as large part load as will ever be required, they will be *small* enough, so that in case of a sudden complete shut down, the water will easily attain its full figured height of y_1 in the riser, and even spill over into the larger vessel. It should always be arranged to do so, at any rate. The method of fixing the size of ports will be pointed out later.

52 As above, we have for acceleration,

$$d\,t = \frac{\frac{L}{g}\,d\,v}{y_1 - C\,(V^2 - V_1^2)}$$

from which, by integration as for (8a) and substitution between the proper limits V_1 and V_2 we have, after simplifying

$$t_a = \frac{L}{2g\sqrt{C\,y_1 + C^2\,V_1^2}} \log_e \frac{(Z - V_1)\,(Z + V_2)}{(Z + V_1)\,(Z - V_2)} \qquad [11]$$

in which Z is substituted for

$$\sqrt{\frac{y_1}{C} + V_1^2}$$

to save repetition.

53 Also, by a process precisely similar to the one for obtaining equation (10) we have

$$y_1 = R\,t_a\,V_2 - \frac{R\,L}{2gC} \log_e \frac{y_1}{y_1 - C\,(V_2^2 - V_1^2)} \qquad [12]$$

in which the same approximate value of y_1 as given for equation (10) may be used as a first trial, and in fact, is often close enough without going further.

54 The size of the ports is easily made consistent with the rest of the design by arranging them of such size as to discharge a quantity of water equal to $A\ (V_2 - V_1)$ under the head of y_1. Within reason-limits there is not much danger of getting them too large and it is always better to err on that side by assuming a value for $(V_2 - V_1)$ fully as large as can ever conceivably be suddenly demanded by an increment of load. $(V_2 - V_1)$ equal to 10 per cent V_2, where V_2 is the full load velocity, is often an outside figure for very large plants, but not for small ones.

55 Thus far not much account has been taken of the swelling action due to the *opening* of the wheel gates during depression of the wave, due to the drop in head, and the *closing* of the gates during surge of the wave due to rise in head. As before stated, this action is unimportant only when the design of the regulator is perfectly adequate in other respects, but it is always best to choose $V_2 - V_1$ large enough by making V_2 large enough to cover the increase in velocity due to the dropping off in head equal to a trial y_1, which would tend to increase V_2 to about $\frac{h\ V_2}{h - y_1}$ where h is the working head. This correction produces quite accurate results in the use of equation (12) and gives fair results, which are on the safe side, in the use of equation (7).

56 When the differential action is omitted, this consideration rapidly increases in importance as the tunnel or conduit becomes longer, and for values of R above a certain critical amount, an inverse function of the length, the waves will accelerate at each vibration, and continue to increase in amplitude until checked by the open top of the surge pipe, after which they will vibrate at constant amplitude indefinitely, so long as the load remains steady.

57 The critical value is difficult to express mathematically, but it evidently may increase as the working head increases and must decrease as L increases; also, it is practically independent of the amount of initial change in load, and independent of the actual velocity values, within working limits.

58 With a proper differential arrangement the value of R could be twice as much as this critical value, and even then it would likely be far on the safe side. This consideration, however, is not of much practical value because almost invariably such a large value of R would make the head variation too great to meet the requirements of good regulation: but without the differential action it is quite

possible to get R too large, especially when the head is low as compared with the length of the conduit. When $\frac{h}{L}$ gets below $\frac{2}{g}$ with L greater than a quarter of a mile or so, it becomes extremely easy to fall into grave errors, if no more thought is given to the size of the standpipe than simply to provide area enough for a moderate initial variation in head.

59 In computing R it must be borne in mind that the area of standpipe which affects these pressure waves is that intersected by a horizontal plane. That is, if the pipe is not vertical it is more effective because the area of the horizontal ellipse is then the governing feature.

POSSIBLE USE OF COMPRESSED AIR

60 When the ground in the immediate vicinity of the power house is low-lying, making it a hardship to carry the regulator structure as high as would be necessary, it is quite feasible and easily practicable to cut down the height by superimposing a false atmosphere of more or less high pressure.

61 This can be done by closing the reservoir at the top and connecting it to a compressed air chamber, providing a fairly ample expansion area. The compressor plant would be insignificant as it would merely have to make up for leaks and absorption of air by the water. The same formulæ apply very well when they are patched up to allow for the expansion and contraction of the air.

62 Before developing equations for the design of the compressed-air regulator, it may be well to point out a new phase of the differential action, as effected by the presence of the internal surge pipe or riser. This pipe is really nothing more nor less than a piezometer, or *measurer* of the accelerating or retarding head acting upon the conduit. Now, this head would exist, whether it were *measured* or not, so we may imagine the riser to be suddenly constricted indefinitely, just above the ports, but having ample entry way from the conduit. Then we may go a step further, and close up the pipe hermetically, just above the ports; and then, since the port holes may as well be horizontal as vertical, we may unite them into one hole at the termination of the short projecting neck of riser.

63 This gradually leads to the use of a short nozzle entering the conduit, and leading, with ample entry way, from the bottom of the tank. Such an affair as naturally designed would make the exit of the water from the tank into the conduit far harder than

entry from the conduit into the tank, and this is precisely what is desired in the operation of a plant, because much larger loads (even full load) are suddenly rejected than can ever commercially be suddenly demanded.

64 Even under atmospheric pressure the riser may sometimes be omitted to advantage, if the constriction at the port or ports makes the operation the same as though it were present, but usually it is better to have it in. The use of compressed air, however, would usually demand its omission, because the vertical dimension of the tank is almost never sufficient to furnish the required room for the water to rise and fall within it.

65 For an ordinary air tank without constriction at the ports we have $\frac{p\,l}{l - y} + y =$ the pressure head, where l is the length or height of the tank *above* the *water level*, or the length of the air column, p is the original steady air pressure above zero, expressed in water-feet, and y is the height of a wave, as before. The net head against which work is done in lifting the water and compressing the air is $\frac{p\,y}{l - y} + y$. Therefore, neglecting friction, we have,

$$\int_o^y \frac{p\,y\,d\,y}{l - y} + \int_o^y y\,d\,y = \frac{L\,R}{g}\int_V^{V_2} (V - V_1)\,d\,v \text{ for retarding.}$$

Integrating and simplifying we have,

$$y^2 - 2\,p\left\{l \log_e\left(1 - \frac{y}{l}\right) + y\right\} = \frac{L\,R}{g}\left\{(V_2 - V_1)^2 - (V - V_1)^2\right\} \quad [13]$$

which becomes, for y_1 which is a maximum when $V = V_1$

$$y_1^2 - 2\,p\,\left\{l \log_e\left(1 - \frac{y}{l}\right) + y\right\} = \frac{L\,R}{g}[\,(V_2 - V_1)^2\,] \quad [14]$$

which holds good for retarding. Similarly, for accelerating we have,

$$y^2 - 2\,p\left\{l \log_e\left(1 + \frac{y}{l}\right) - y\right\} = \frac{L\,R}{g}[\,(V_2 - V_1)^2 - (V_2 - V)^2\,] \quad [15]$$

66 The time of vibration and the introduction of friction may lead to results by processes similar to those used in the previous solutions, but inasmuch as friction is often negligible where compressed air is used, and the period of vibration has little practical

value, the writer has not taken the time to go into these tedious details. Besides, the differential action *should never be omitted.*

67 Let us now study this action under compressed air, and let b be the ratio of the instantaneous rise of pressure in the *imaginary* or *existing* internal surge pipe, to the ultimate maximum height of water wave, which we shall still designate as y_1. Let the value of R be such, that the ultimate rise of water to a value y_1 plus the additional pressure due to compressing the air will just equal the first temporary instantaneous rise of pressure which we are expressing as $b\,y_1$. If the value of R is eventually going to be fixed as above described, it follows that the retarding or accelerating pressure *level* will hold practically constant during the period of retardation or acceleration, and will have a value of $b\,y_1$ and the accelerating or retarding head will have a value of $b\,y_1 - Z$ where Z, as before, represents the change in hydraulic gradient.

68 The area of the ports to meet this condition must be

$$a = \frac{A\ (V_2 - V_1)}{\theta\sqrt{2\,g\,b\,y_1}}$$

where θ is the coefficient of discharge. Furthermore, inasmuch as we may have V maximum suddenly rejected and we may make the coefficient of nozzle discharge different, to suit, in exit and entry of water, it should be designed for two coefficients, θ' and θ'' for accelerating and retarding respectively, and we should have

$$a = \frac{A\,(V_2 - V_1)}{\theta'\sqrt{2\,g\,b\,y_a}} = \frac{A\,V_{max}}{\theta''\sqrt{2\,g\,b\,y_r}} \qquad [15a]$$

where $V_2 - V_1$ represents a large load suddenly demanded and V maximum represents full load suddenly rejected. Referring to equation (11) and putting $y_1 = b\,y_1$ we have,

$$t_a = \frac{L}{2\,g\sqrt{C\,b\,y_1 + C^2\,V_1^2}}\log_e\frac{(Z^1 - V_1)\,(Z^1 + V_2)}{(Z^1 + V_1)\,(Z^1 - V_2)} \qquad [16]$$

in which

$$Z^1 = \sqrt{\frac{b\,y_1}{C} + V_1^2}$$

similarly to (11). Comparing equation (12) and introducing by_1 for y_1 in the right hand side of the equation, we have,

$$y_1 = R\,t_a\,V_2 - \frac{R\,L}{2gC}\log_e\frac{b\,y_1}{b\,y_1 - C\,(V_2^2 - V_1^2)} \qquad [17]$$

from which R may be found with a trial value for y_1 and b. The value of b has to be selected according to judgment, and is fixed entirely by the proportions of the air chamber and the air pressure as follows: According to the original assumption of the value of b,

$$\frac{p\,l}{l \mp y_1} \pm y_1 = \pm b\,y_1 + p$$

from which,

$$b = \frac{p}{l \mp y_1} + 1 \qquad [18]$$

and

$$l = \frac{p}{b - 1} \pm y_1 \qquad [19]$$

69 Note that b has slightly different values in accelerating and retarding due to change of sign. If y_1 and l are selected by judgment, then b follows, and R may be determined from (16) and (17). If the proportions do not suit, a new trial must be made, as there is probably no easy way to put these things together.

70 The decrease or variation in head on the water wheels, which is the thing usually desired to limit, is naturally $b\,y_1$ or, as above, for acceleration $\frac{-p\,l}{l + y_1} + y_1 + p$. So that b, l, and y_1 should really be selected together according to judgment before attempting to work out t_a or R. Similarly, for retarding (cf. equations (9) and (10))

$$t_r = \frac{L}{g\sqrt{C\,b\,y_1 - C^2\,V_2^2}}\left\{\tan^{-1}\frac{V_2}{\sqrt{\frac{b\,y_1}{C} - V_2^2}} - \tan^{-1}\frac{V_1}{\sqrt{\frac{b\,y_1}{C} - V_2^2}}\right\} \qquad [20]$$

and
$$y_1 = \frac{R\,L}{2gC}\log_e\frac{by_1}{b\,y_1 - C\,(V_2^2 - V_1^2)} - R\,t_r\,V_1 \qquad [21]$$

and for a complete shut down where $V_1 = 0$ and $V_2 = V$ maximum we have,

$$y_r = \frac{R\,L}{2gC}\log_e\frac{b\,y_r}{b\,y_r - C\,V^2_{max}} \qquad [22]$$

in which y_r may be found by successive approximations, and when

it is found, the value of θ'' may be determined from equation (15*a*), θ' having been determined from the assumed value of y_1 for accelerating. It is not absolutely necessary to have two values for θ, but this method is a desirable refinement. In presenting these formulæ for the design of regulators, the writer desires to point out very clearly that the equations have not had the benefit of a thorough independent check—they are written rather hurriedly and if not absolutely right, as it is believed they are, the methods at least are correct and nothing more than a slip in algebraic transposition, or an error in integration, will likely be found. It is hoped that this presentation will call out all the necessary critics and checkers and clear the subject of all useless or erroneous matter.

71 The regulation is nearly as good with the air as without it, if the differential feature is adhered to, but otherwise not. In such a case, the advantage of this feature is greater even than in the ordinary case.

72 The accompanying curves illustrate the difference in shape of the pressure wave between that produced in an ordinary open tank, and that produced under the same conditions when a small surge pipe is inserted within it. The periods are not quite the same on account of the reduction in the reservoir area due to the presence of the internal pipe. The quantities and constants assumed in working out these curves are as follows:

$A = 220$	The diameter of surge pipe = 9 ft.
$R = 0.112$	" " " tank = 50 ft.
$L = 15200$	Aggregate area of ports = 20 sq. ft.
$V_2 = 11$	
$V_1 = 9.8$	
$h = 540$	

Friction head 20 ft. for an 11 ft. velocity.

The action of the wheel gates is taken into consideration.

73 It may help give a practical flavor to this discussion to know that these particular curves represent conditions which would exist in the Feather River, Cal., development at Big Bend, in following out a design for a surge tank regulator suggested by John R. Freeman, C.E., and approved with some minor modifications by Viele, Blackwell & Buck of New York.

74 Where it is desired to catch all the water under all conditions, the internal surge pipe has a particularly advantageous feature in

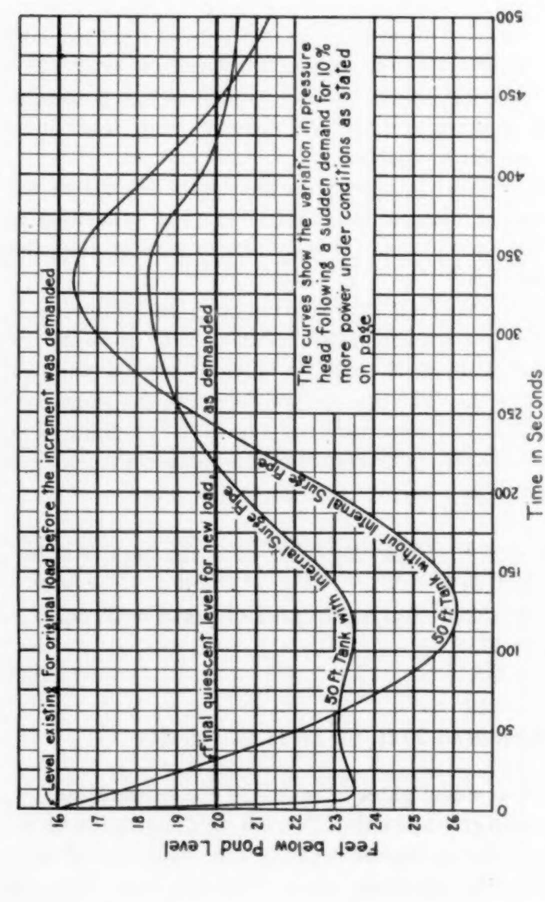


FIG. 1 COMPARATIVE PRESSURE WAVES IN OPEN TANK AND IN TANK WITH SMALL SURGE PIPE

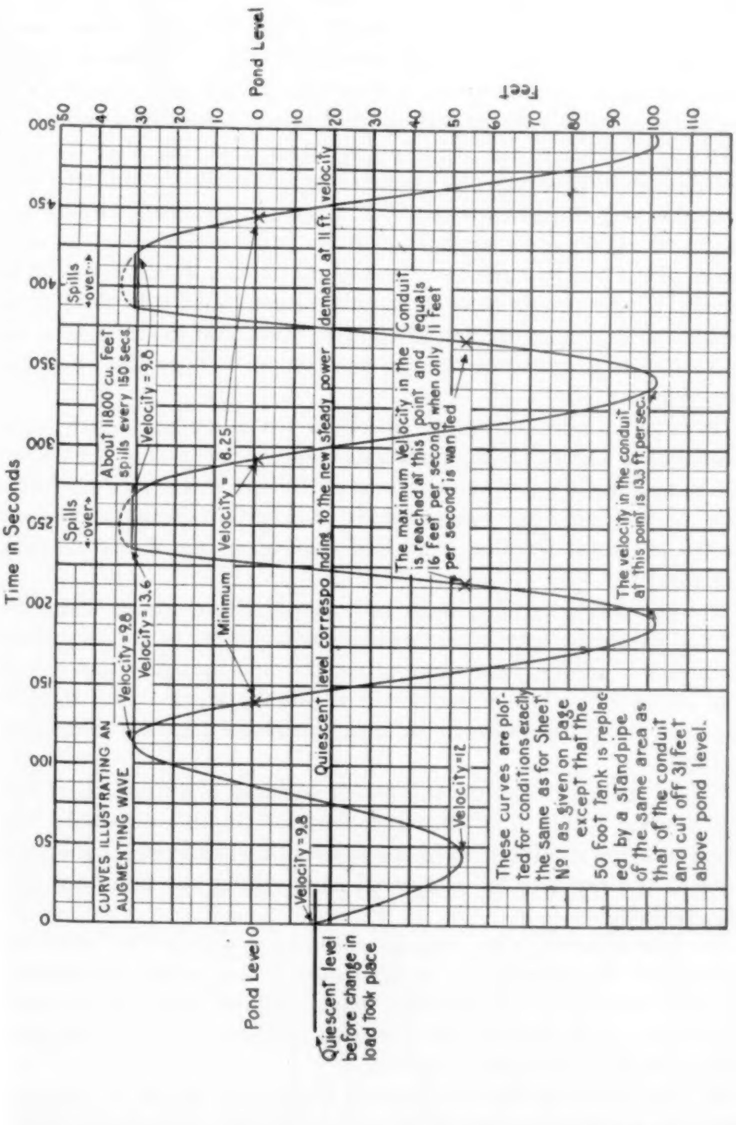


FIG. 2 PRESSURE WAVE WHERE STAND PIPE IS USED

permitting an effective factor of safety by simply extending it above the top of the tank, although the tank may have been figured not to overflow with the surge pipe cut off level with it. When the tank is not made ample to catch all the water in case of a shut down, some provision must naturally be made to carry it away. In case the differential action is used, however, it is usually just as cheap to catch all the water as to provide for getting rid of it, and a good deal neater.

75 The writer would like to point out emphatically that the subject matter of this paper is intended to treat only of the long and comparatively sluggish waves which vibrate in the main conduit as a unit, and *must not be confused with the rapid waves which serve to communicate pressure from one end of the column to the other.* The velocity of these latter waves is so great as compared with the time of oscillation of the surge waves, in a long conduit, that the time occupied in transmitting pressures through the column is neglected in all the foregoing studies.

76 Furthermore, adequate provision for damping these surges, according to the foregoing formulae, does not mean that regulation will be satisfactory *if the closed penstock between this regulator and the power house is too long or carries water at too great velocities* in proportion to the total working head and kinetic energy of the rotating machinery.

77 The closed turbine feeder or penstock also contains within itself two distinct classes of wave phenomena similar to the long main conduit, but in this case there is usually not enough difference in time between the oscillating period of the surge wave and time of travel of the pressure wave so that the latter may be altogether neglected.

78 In the long conduit, the period of surge is a function of the length alone where R is constant; whereas, in the penstock, the period of surge is a function of the speed of the waterwheel governor which is too rapid to be considered as anything but infinite in the long conduit.

79 It is apparent then, that at least two considerations make the treatment of the closed penstock different from that of the long conduit, viz., the velocity of the pressure wave and the speed of governor travel which in the latter case are both regarded as infinite as compared with time of surge vibration.

80 The intimate relation between the kinetic energy in the penstock, the stroke of the governor, and the kinetic energy in the rotating machinery is not studied in this paper, as it would require

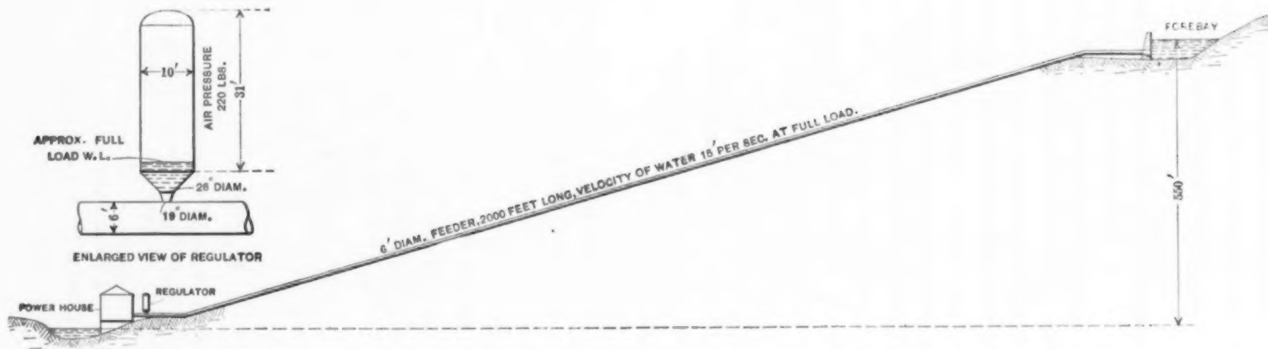


FIG. 3

Diagram showing the comparative dimensions of a compressed air differential regulator which will effect ideal control of the assumed 2000-ft. water column making the speed regulation of the water wheels commercially perfect under all conditions, by throttling the stream, without wasting any water. 15 per cent increments of load may be instantaneously demanded and almost instantly furnished by the governor, with less than 10 per cent drop in pressure and no appreciable speed variation, even though the fly wheel effect of the rotating parts is small.

more time to set forth than is at present available for the purpose; but the writer would like to offer an empirical rule which is usually on the safe side, and often safer than is necessary, as follows:[1]

81 If the length of the penstock is not greater than 18 times the head divided by the velocity, no difficulty will be experienced in regulation, unless the load changes are abnormal as compared with the average plant. For very high heads where the wheel speed is likely to be high, this expression sometimes becomes altogether too safe but at any rate it may serve someone as a rough check.

82 If the conduit is, for any reason, extended beyond the penstock connections, before coming to the regulator, as is often desirable, then the effective length of penstock is increased, so far as regulation is concerned, and should be considered equal to $L' + R' L''$, where L' is the actual length of penstock, R' is the ratio of the aggregate penstock area to the area of the extension section, and L'' is the length of such extension.

83 The writer will close this paper with one or two numerical examples illustrating the application of his formulæ.

84 Let us imagine a plant of following dimensions:

$$L = 5000 \text{ ft.}$$

$$A = 100 \text{ sq. ft.}$$

$V_{max.} = 15$ ft. per second = maximum draft velocity at lowest point of surge and is substituted for V_2 in formula according to Par. 55.

$$C = \tfrac{1}{10}$$

$$h = 200 \text{ ft.}$$

The market or use for the power, is such, we will say, that we are likely to have sudden changes in load requiring an increment in velocity of, say, 2 ft. per second, from 13 ft. to 15 ft. per second when the opening of the wheel gates and the change in effective head are considered. This amount or rate of change is a matter of judgment and must be estimated large enough for any particular plant and market according to the special requirements of the case, always remembering that

[1] The article by Irving P. Church in the Transactions of the Association of Civil Engineers of Cornell University, 1898, is of assistance in this study, but has not been carried out so far as to entwine into the equations the effect of the momentum of the rotating machinery as well as that of the moving water column, and hence is still incomplete for correct application to power plant studies, although very novel and useful for the scope intended by Professor Church.

larger part loads are likely to be rejected than demanded, except just after an accidental shut down, when the reverse may be the case.

85 In selecting the increment in velocity it is better to use the high actual velocity values as giving slightly more unstable conditions. C should be chosen high enough to cover future deterioration of the smoothness, if necessary.

86 For this change in velocity of 2 ft. per second let it be required to keep the variation of head within 5 per cent, or say, not more than 10 ft., and in the first place let us figure the size and height of tank without any differential action. In equation (7) putting $y_{max.} = 10$ and solving for R we have,

$$100 = \frac{5000\ R}{32.2}\,(15 - 13)^2 + 0.01\ (15^2 - 13^2)^2 \text{ or } R = \text{about } 0.11$$

Hence the proper area of tank $= \dfrac{A}{R} = 910$ or diameter $= 34$ ft.

87 Now let us see what the surge would be in case of a shut down. To do this, put $V_2 = 15$ and $V_1 = 0$ in the same formula, and solve for $y_{max.}$ thus:

$$y^2_{max} = \frac{5000 \times 0.11}{32.2}\,(15)^2 + 0.01 \times 15^4 \text{ or } y_{max} = 66 \text{ ft.}$$

which is to say that the water will rise 66 ft. above its existing full load level, which latter is $C\ V^2$ *below* the pond level, or $\frac{1}{10} \times 15^2 =$ 22.5 ft., hence the elevation of the top of the tank above the pond level must be 66 − 22.5 or 43.5 ft. if it is desired to catch all the water after a shut down. This dimension will be on the safe side because all the water is never permanently rejected at the wheels, enough being retained to keep the runners up to speed. For this consideration it is obvious that a value of C should be chosen as low at least as it is likely to be in practice.

88 The total height of the tank is then, theoretically, 10 ft. below the 13-ft. velocity gradient + 43.5 ft. above pond level or $10 + \frac{1}{10} \times 13^2 + 43.5 = 70.4$ ft. In case of a complete shut down, regulation is of no consequence, and care should be taken only that the pressures do not become unsafe. Let us now compare the dimensions of a tank containing an internal surge pipe 6 ft. or 7 ft. in diameter. The writer has found that within all reasonable limits the size of this pipe does not make much difference. It should be small for two reasons, economy and because with a small pipe the level takes its new position quickly and hence more nearly conforms to the theory

upon which the tank dimensions are computed. It should be large enough so that the choking action due to it will not cause undue rise in pressure.

89 Reverting to formula (11) with $y = 10$ ft., $(V_2 - V_1) = (15 - 13)$, etc., let us solve for t_a or the time occupied in accelerating the water in the conduit from 13 ft. to 15 ft. per second, while the water in the surge pipe maintains a steady level 10 ft. below the end of the 13-ft. velocity gradient. This condition is very nearly brought about, as before pointed out, by giving such a size to the ports that they will discharge A (15 − 13) cubic feet per second under the head y or 10 ft. Therefore, if the coefficient of discharge is, say 0.6, we shall have the aggregate area of the ports,

$$a = \frac{A\,(V_2 - V_1)}{0.6\,\sqrt{2\,g\,y}} = \frac{100 \times 2}{0.6\,\sqrt{2\,g \times 10}} = 13.2 \text{ sq. ft.}$$

The ports may be a little smaller, and quite a lot larger without having much of any effect upon $y_{max.}$ In general, it is well, perhaps, to add some 5 per cent to the values thus obtained. Furthermore, the coefficient of discharge should be carefully considered according to the nature of the opening, the approach and exit of the water, and should be chosen as low as it is likely to be, as it is rather better to err on the side of too large "a" than too small.

90 To resume formula (11) we have

$$t_a = \frac{5000 \times 2.3}{64.4\,\sqrt{1 + 1.69}}\ \text{com. log}\ \frac{(16.4 - 13)\ (16.4 + 15)}{(16.4 + 13)\ (16.4 - 15)} = 45.1 \text{ sec.}$$

Now substituting this value in equation (12) we may solve for R thus

$$10 = R \times 45.1 \times 15 - \frac{R \times 5000 \times 2.3}{3.22}\ \text{com. log} \sqrt{\frac{10}{10 - 0.1\,(\overline{15^2 - 13^2})}}$$

or, $R = 0.244$. The net tank area is therefore $\frac{100}{0.244}$ or 410 sq. ft. as against 910 without the differential action. (These computations were made entirely on the slide rule and are not particularly close). We have therefore made a saving of $\frac{5}{9}$ plus the additional small cost of the riser, without taking into account any change in height, which we will now calculate. In order to catch all the water after a shut down let us revert to equation (10) and figure $y_{max.}$ with $R = 0.244$, $V_2 = 15$, $V_1 = 0$, etc. It will be noticed that the latter term vanishes and we have,

$$y = \frac{0.244 \times 5000 \times 2.3}{3.22}\ \text{com. log} \sqrt{\frac{y}{y - 0.1\,(15)^2}}$$

This must be solved by trial, and as a first approximation solve for y in equation (7) as follows, putting $R = \frac{1}{2} R$:

$$y^2 = \frac{5000 \times 0.244}{64.4} (15)^2 + 0.01 (15)^4 = \text{or } y = 69 \text{ feet.}$$

Substituting this above we have, $y = 872$ com. log $\sqrt{1.485} = 74.9$ from which we see that the trial value of y was too small so we now try $y = 71.5$ and we have $y = 872$ com. log $\sqrt{1.459} = 71.5$ which agrees with our assumption and is the correct value. The total height of tank and riser would be, as before $10 - \frac{1}{10} (15^2 - 13^2) + 71.5 = 75.9$.

91 In order to be sure that we have not carried the riser too high in proportion to the size of the ports we must see that their area is small enough so that water will be sure to reach at least the top of the riser. It is preferable to have it spill over into the tank which it will be sure to do if a is well under $\dfrac{AV_2}{0.6 \sqrt{2gy}}$ or when $a = 13.2 < \dfrac{100 \times 15}{0.6 \sqrt{64.4 \times 71.5}} = 36.8$, so in this case, as usually, spilling will be bound to take place, and our formula for $y = 71.5$ ft. is on the safe side, for we have neglected the friction and choking heads of the high velocities in the riser due to shutting down full load, and disregarded the water necessary to hold the wheels at speed.

92 To sum up then, we can get better regulation and provision for catching energized water, with a tank 22.8 ft. diameter $\times$ 75.9 ft. high and a riser 7 ft. diameter $\times$ 75.9 ft. high than we can with a single tank 34 ft. diameter $\times$ 70.5 ft. high which would cost nearly twice as much.

93 The differential regulator is better aside from the cost consideration, as already pointed out, because it has an increased damping effect upon the surge waves after their inception, and renders less likely a troublesome condition due to coincidence of waves which would result from load changes occurring at critical times in such manner as to superimpose a wave upon one which was not yet dead.

94 In conclusion the writer would like to say that up to date he has had the benefit of very little criticism of his methods of treating these problems and is presenting this paper in the hope that it will interest a number of people sufficiently so that they will help to correct any errors which he may have, singlehanded, fallen into,

and to expand and improve the mathematical treatment which has been developed without a particularly sound knowledge of differential equations.[1]

DISCUSSION

MR. L. F. HARZA I am interested in Mr. Johnson's paper, for the reason that I have spent considerable time during the past winter studying the same problem. I was, at that time, asked by Prof. Daniel W. Mead to make a mathematical study of the general problem of speed regulation of water wheels, to see if some much needed light could not be thrown upon this subject that would make the solution of speed regulation problems depend upon scientific analysis.

2 The studies were made for Professor Mead, and at his expense, for his forthcoming work, Water Power Engineering.

3 The problem of the standpipe arose in connection with the general problem of speed regulation, and the results accomplished were quite gratifying. So far as I knew at that time no other solution of the problem had ever been made. Apparatus has been built, and experiments are now being conducted by Mr. A. H. Ayers in the Hydraulic Laboratory of the University of Wisconsin to verify the formulae experimentally.

4 The portion of the paper relating to the differential regulator or the surge pipe and its use, is entirely new to me. Most of the equations, however, given by Mr. Johnson as applying to the simple standpipe, and some additional ones, were derived by the writer by a different method, and two of his equations, [3] and [5], were obtained in much simpler form. The method used was as follows:

Let A = cross-sectional area of the penstock.
F = cross-sectional area of stand pipe.
q = water used by wheel.
v = penstock velocity.
y = instantaneous elevation of water in the standpipe above or below that in the forebay, or the "surge."

[1]Patents have been applied for, covering the differential feature of the author's device, because the correct use of it requires careful study and it is hoped that a certain amount of restraint in its use will serve better to control its proper application, and render less likely costly misuses which might reflect discreditably upon the whole idea.

Y = maximum surge, neglecting friction and governor action.

T = time for return to normal level, neglecting friction and governor action.

T' = time for return to normal level, friction and governor action considered.

H = total head.

h = instantaneous head.

D_a = maximum upward surge when full load is rejected.

D_b = maximum downward surge below initial friction gradient.

D'_b = maximum downward surge below initial friction gradient with effect of friction omitted.

D_B = maximum downward surge below forebay level.

5 Now, it can readily be shown that

$$\frac{dv}{dt} = \frac{g}{L} \text{ (accelerating head)}$$

$$= \frac{g}{L} (y - \text{friction head in penstock})$$

Or

$$\frac{dv}{dt} = \frac{g}{L} (y - c\, v^2) \qquad [1]$$

It is evident that

$$\frac{dy}{dt} = \frac{A\, v - q}{F} \qquad [2]$$

6 If the governor keeps step with the change in head and maintains a constant power, then

$$q\, h = q_2 h_2$$

$$q\,(H - y) = A\, v_2\,(H - c\, v_2^2) \qquad [3]$$

7 Equation [2] now becomes:

$$\frac{dy}{dt} = \frac{A}{F}\left[v - \frac{v_2\,(H - c\, v_2^2)}{(H - y)}\right] \qquad [4]$$

8 Differential equations [1] and [4] form a complete expression of the existing relations, but are very difficult, if not impossible, to integrate mathematically. The integration has, however, been per-

formed for several problems by the successive application of the equations to small portions of the arc as described by Mr. Johnson, obtaining a damped harmonic similar to what he describes.

9 Attempts were also made by the writer to integrate equation [1] with the cv^2 term omitted, simultaneously with equation [4], thus to obtain the effect of governor action alone, and to integrate equation [1] together with [2], where q is constant and equal to Av_2, in order to find the curve resulting from friction alone. Both attempts have failed thus far.

10 If both friction and governor action are neglected, we have

$$\frac{dv}{dt} = \frac{g}{L}\, y \qquad [5]$$

$$\frac{dy}{dt} = \frac{A}{F}\,(v - v_2) \qquad [6]$$

11 The writer found these equations to resemble very closely the simple sine curve when plotted by the arithmetical method, and therefore assumed the equation of the general sine curve, differentiated it, and equated the derivatives to those given by [5] and [6] for certain determining points of the curve. By this method, expressions were readily derived for Y and T (one-half the wave cycle) and the following v–t and y–t equations obtained. Later on, the writer succeeded in integrating equations [5] and [6] mathematically, and obtained the same identical equations as before:

$$T = \pi \sqrt{\frac{F\,L}{A\,g}} \qquad [7]$$

$$Y = \pm \sqrt{\frac{A\,L}{F\,g}}\;(v_2 - v_1) \qquad [8]$$

$$y = Y \text{ sine } \frac{\pi}{T}\, t \qquad [9]$$

$$v = v_2 - (v_2 - v_1) \cos \frac{\pi}{T}\, t \qquad [10]$$

12 In order to obtain more valuable equations, which would include the effect of governor action and friction, the following method was used:

13 . Let the time required to reach D_b and hence to reach approximately the value v_2, under exact conditions, be $\frac{T'}{2}$.

14 The time $\frac{T'}{2}$ will be slightly greater than $\frac{T}{2}$ when friction and governor action are involved, and the method of determining it will be given later (equation [22]).

15 It is evident that the number of foot pounds of energy which must be supplied by the standpipe in this time $\frac{T}{2}$ is equal to the energy required by the wheel plus that required to accelerate the water in the penstock plus that necessary to overcome the friction of the penstock minus that supplied through the penstock,

Or

$$E_s = E_w + E_a + E_f - E_p \qquad [11]$$

16 Now

$$E_s = w\,F\,D_b\left(H - cv_1^2 - \frac{D_b}{2}\right) \qquad [12]$$

where D_b is the maximum surge below the initial friction gradient for v_1, and is used in place of Y to distinguish it from the value obtained by the other foimula.

17 Also

$$E_w = A\ v_2\ \frac{T'}{2}\ w\ (H - cv_2^2) \qquad [13]$$

18 And

$$E_a = \frac{w}{2\,g}\ A\ L\ (v_2^2 - v_1^2) \qquad [14]$$

19 To obtain E_f we have

$$d\,E_f = A\,v\,w \times cv^2\,dt \qquad [15]$$

where c is the friction coefficient and v is obtained from equation [10].

$$v = v_2 - (v_2 - v_1)\cos\frac{\pi}{T}\,t$$

20 The integration of [15] between the limits $t = \frac{T'}{2}$ and o, gives

$$E_f = A\, w\, c \left[\frac{v_2^3\, T'}{2} - \frac{3\, T'}{\pi}\, v_2^2\, (v_2 - v_1) + \tfrac{3}{4}\, T'\, v_2\, (v_2 - v_1)^2 - \frac{6T'}{e} \right.$$

$$\left. (v_2 - v_1)^3 \right] \qquad [16]$$

21 Also to find E_p we have

$$d\, E_p = H\, A\, w\, v\, dt,$$

where v is obtained from equation [10] as before. Integrating between the limits $\frac{T'}{2}$ and o, gives

$$E_p = H\, A\, w\, T' \left(\frac{v_2}{2} - \frac{v_2 - v_1}{\pi} \right) \qquad [17]$$

22 Combining and simplifying:

$$D_b^2 - 2\,(H - c v_1^2)\, D_b = - \frac{2\, A}{F} \left\{ \frac{L}{2\, g}\, (v_2^2 - v_1^2) + C \left[- \frac{3\, T'}{\pi}\, v_2^2 \right. \right.$$

$$\left. (v_2 - v_1) + \tfrac{3}{4}\, T'\, v_2\, (v_2 - v_1)^2 - \frac{T'}{6}\, (v_2 - v_1)^3 \right]$$

$$\left. + \frac{H\, T'}{\pi}\, (v_2 - v_1) \right\} \qquad [18]$$

$$D_B = D_b + c v_1^2 \qquad [19]$$

23 The upward surge can be found by the same equation by a proper change of signs, but is unimportant since it is always less than the downward surge D_B for the same change of velocities.

24 If friction be omitted and T' be changed to T for reasons mentioned later, equation [18] reduces to

$$D'^2_b - 2\, H\, D'_b = - \frac{2\, A}{F} \left\{ \frac{L}{2\, g}\, (v_2^2 - v_1^2) + \frac{H\, T}{\pi}\, (v_2 - v_1) \right\} \qquad [20]$$

25 To derive an equation for the maximum upward surge D when full load is rejected, we may equate the original kinetic energy in the penstock to that expended in friction plus that used in raising

water in the standpipe. The energy lost in friction is found from equation [16] by putting $v_2 = o$, or

$$E_f = \frac{A\,w\,c\,T\,v_1^3}{6}$$

26 The other quantities are evident. This gives·

$$\frac{W\,A\,L}{2\,g}\,v_1^2 = \frac{A\,w\,c\,T\,v_1^3}{6} + \frac{w\,F\,D_a^2}{2}$$

Or

$$D_a^2 = \frac{A}{F}\,v_1^2\left(\frac{L}{g} - \frac{c\,T\,v_1}{3}\right) \qquad [21]$$

27 Equations [18], [19], [20] and [21] are all theoretically exact, except for the assumption that the velocity-change takes place along a simple harmonic in time $\frac{T}{2}$. The true curve for a half cycle, as used, is scarcely distinguishable from a simple harmonic but its period or time for return of water in standpipe to normal level is greater than the value T, given by equation [7]. In two cases which the writer has solved by arithmetical integration and in the example given by Mr. Johnson of the conditions at the Feather River Plant, shown in Fig. 2, the true value T' may be closely approximated by the following formula:

$$T' = \frac{D'_b}{Y}\,T \qquad [22]$$

where T is found from equation [7], Y from equation [8], and D'_b from equation [20].

28 The quantity T' is useful in itself as the true time for return to normal head, but its use in formula [18] for determining D_b is not advisable, as the writer has found by solving a number of problems that the value of D_b thus found agrees almost exactly with the value of D'_b found from equation [20], in which equation the value of T from equation [7] is used. Equation [18] may thus be rejected entirely, and equation [19] becomes

$$D_B = c\,v_1^2 + D'_b \qquad [23]$$

29 For a basis of judging the accuracy of approximate formulæ three problems have been selected, all of which have been solved by arithmetical integration, and whose correct solutions are therefore known. Problems 1 and 2 have been solved by the writer, and the solution by arithmetical integration of Problem 3 has been given by Mr. Johnson in Fig. 2.

	H	L	A	F	R	v_1	v_2	C
Problem 1......	50	500	50.3	402.4	0.125	1.94	4.77	0.03
Problem 2......	200	5 000	100	910	0.11	13	15	0.1
Problem 3	540	15 200	220	1964	0.112	9.8	11	0.1654

30 The degree of accuracy obtained in applying equation [22] is shown by the following table:

	T' (True value by arithmetical integration.	T' by equation [22]
Problem 1....................	40.6	40.4
Problem 2....................	153	156
Problem 3....................	240	234

31 The agreement of formulæ ([18] and [19]) and ([20] and [23]) for D_B is shown below:

	D_B by equation [19]	D_B by equation [23]
Problem 1....................		4.66
Problem 2....................	28.2	28.85
Problem 3....................	26.7	26.0

32 This evidently indicates that equations [18] and [19] may be discarded.

33 The following comparison of the results of the writer's equation for D_B with Mr. Johnson's equation, and with the correct values, shows a remarkable accuracy of both methods.

	cv_1^2	By arithmetical integration	Johnson's equa. [7] $+ cv_1^2$	Writer's equation [23]
Problem 1............	0.11	4.75	4.45	4.66
Problem 2............	16.9	28.15	26.9	28.85
Problem 3............	16.0	26.1	25.66	26.0

34 If full load ($v = 15$) be suddenly thrown off in problem 2, Johnson finds the maximum upward surge to be 43.5 ft. above pond level. The writer's equation [21] gives 48.8 ft., and the correct value obtained by arithmetical integration with one-second intervals is 47.8 ft. This is the only problem which the writer has solved to check these formulæ.

35 The above comparison shows that both methods for determining D_B give results very close to the truth, and much closer than the accuracy which can be obtained in estimating the maximum instantaneous load change which should be provided for. The danger of the piling up of waves also adds an additional factor which must be left to the judgment of the engineer.

36 In the opinion of the writer, the standpipe should not be built high enough to hold all the water when full, or nearly full, load is rejected by the wheels, but should preferably be built to overflow either at the top or through relief valves at the bottom set to waste when the water reaches a given height.

37 This overflow provision not only limits the upward surge but also thereby limits the maximum possible downward surge which can occur, and prevents the possibility of the piling up of waves above a certain value, which can be definitely figured, and depends upon the height of overflow above the forebay. The water which would waste at the rate intervals when full load is rejected would be of slight consequence, and the large additional height of standpipe, which would in many cases be necessary to conserve this water, would not be economical. For example, in problem 2 the standpipe would need to be built about 50 ft. above the forebay. The higher the standpipe is built, the less often will it overflow, and economy of design would call for a proper balance between first cost and water economy. In addition to economy, the overflow serves to damp the surge and prevent the piling up of waves.

38 Mr. Johnson has based his original equations for the differential regulator upon some assumptions, the truth of which must be demonstrated by arithmetical integration or by experiment.

39 The writer has not yet had time to apply arithmetical integration to the differential features, but as he claims to have done so many times himself, the form of pressure curve which he describes is doubtless correct. If the newly required velocity is to be obtained in approximately the same length of time as with a simple standpipe, it is evident that the average accelerating heads in both cases must be equal. Therefore, if the differential feature produces approxi-

mately a constant accelerating head, this head, and hence the surge, can be less than with the simple standpipe in about the ratio of the average ordinate of the sine wave to its maximum ordinate, or $\frac{2}{\pi} = 0.637$. The differential feature doubtless changes somewhat the time required for this change of velocity, and hence also the relative amplitude of the surge waves.

40 Assuming that the surge is reduced by this differential feature, some obvious advantages result, as follows:

a When the unit has been working at part load, and full load is suddenly demanded, the reduction of the surge evidently helps to prevent the effective head from dropping to such a point that the power demanded cannot be delivered.

b The surge waves are made to die out more rapidly than with a simple standpipe, without overflow, which makes less likely the dangerous piling up of the waves from several successive gate movements.

41 To determine the general effect of the differential feature upon speed regulation, let us consider the duties of the governor. The speed of a unit will remain normal as long as the power output of the wheel equals the demand. This power output is represented by the quantity qh, where q is the actual water passing through the wheel, and h is the effective power head. The function of the governor is to maintain this power product at a value just equal to the demand, by readjusting q for each change which may he made in the demand for power, or which may occur in the head, h.

42 Now, three influences oppose speed regulation:

a The governor cannot act until a change of speed has occurred, of a magnitude depending upon the sensitiveness of the governor. During this interval the wheels have been developing the original power corresponding to v_1, while the generator has been delivering the power corresponding to v_2. The result is the loss of, or absorption of, energy by the rotating parts, which results in a change of speed.

b After the governor has begun to move the gates, some time is required to complete the movement during which time the speed departs still further from normal.

c After the gates have reached their proper position, an appreciable time is required for the water in the penstock

to accelerate up to the newly required value. This time varies from zero, for a zero length of penstock and draft tube, to a period of many seconds for a long closed penstock.

43 The penstock should be measured, for comparing speed regulation conditions, from forebay or standpipe if one is used to tail water.

44 Now, if the turbines were to take their water directly from a simple standpipe, without appreciable length of penstock, then the first and second influences are the only ones opposing speed regulation. If we assume after the initial gate movement resulting from a load change that the speed returns to normal, then it will not again depart from normal except by an amount sufficient to actuate the governor. The slow wave-like variations which take place in h can be readily compensated for by the governor, without the sudden large gate movement which was at first here required to compensate for the load change. Poor speed regulation results from quick changes of effective head, rather than from large changes (within reasonable limits), and a small quick change in h may cause a larger speed variation than a much larger slow change.

45 In the case of the differential regulator the sudden increase of load is followed, as the gates move, by the sudden drop in effective head in the surge pipe, which in the standpipe, did not occur to an appreciable extent until the governor had made the gate adjustment for the load change, and even then took the form of a gradual instead of a sudden drop. Thus the surge pipe, in effect, adds to the load for which the governor must compensate. As the power of a wheel is proportional to the three halves power of the head, it follows that a sudden drop of 5 per cent in the effective head on a wheel would be equivalent in its demand upon the governor and therefore in its effect upon speed regulation to a sudden increase of load amounting to

$$\left(\frac{100}{95}\right)^{3/2} = 1.08$$

or 8 per cent. This must be added to the real load-increase to get the increase which is effective so far as speed regulation is concerned.

46 If there is a considerable length of penstock between the surge tank and the wheel, then the bad effect upon speed regulation is augmented. This is due to the fact that a smaller head than in a simple standpipe becomes available for accelerating the water in this portion of the penstock. A greater time is required to generate the

newly required penstock velocity under this reduced head, with a consequent deficiency of developed power and drop in speed of the unit.

47 The value which is placed upon speed regulation by different engineers and the attention which is given to this problem in the design of plants differ widely. It is now, and must always remain, a matter of individual judgment as to how much in any particular instance can be spent to obtain close speed regulation. The effect of the surge pipe might not render the speed regulation unsatisfactory if the other factors controlling speed regulation were favorable.

48 The writer does not wish it understood that he is opposing entirely Mr. Johnson's differential scheme or trying to underestimate the value of the principle which he has discovered and developed. A knowledge of the exact effect of a resistance between penstock and standpipe is of much value, and much credit is due Mr. Johnson for his thorough analysis of the problem.

49 The writer believes that Mr. Johnson's criticism of Mr. H. E. Warren's able paper upon Speed Regulation of High Head Water Wheels is unwarranted. The governor-controlled by-pass valve which Mr. Warren claims to be "the only complete remedy for the troubles in speed regulation caused by excessive inertia of a water column," is, in reality, the only scheme except the deflector nozzle, which can vary the power derived from a long penstock without change of effective head, and is therefore the most perfect scheme as regards speed regulation alone. Water economy, whether it results from the use of fly wheel, standpipe, or surge tank, brings with it speed regulation troubles, more or less severe, depending upon the natural conditions as well as upon the care and expense exercised in the design and construction of the plant.

PROF. I. P. CHURCH[1] The subject treated in this valuable paper is of such importance as to justify the devotion of a great deal of time and attention on the part of hydraulic engineers and others. For his own part the present writer has been much interested in deriving numerical results in a few specific cases to ascertain the degree of approximation attained in the use of an important formula of the paper; viz: equation [7] of Par. 45. The laborious processes necessary to get anything approaching accurate results have taken so much of the

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writer's time, however, that he is obliged to limit this discussion to the consideration of the single relation referred to, and a presentation of the numerical results obtained from various data.

2 The fundamental relations between the quantities involved in the problem, as derived from the principles of mechanics, will first be presented. Fig. *A* shows, in a diagrammatic way, a vertical section of the long pipe, or main conduit *E X*, and standpipe (or "surge tank") *F W*, etc., of a water power plant. As to notation, let *L* denote the length, *D* the diameter, and *A* the sectional area (i.e., $A = \pi D^2/4$), of the main conduit. The sectional area of the standpipe is A/R, *R* being a ratio. For simplicity it will be supposed that the motor employed is an impulse wheel of the Pelton type, actuated by a "free jet" issuing from the nozzle at *r*; the sectional area of this jet being A'', and velocity V''. When normal steady flow is proceed-

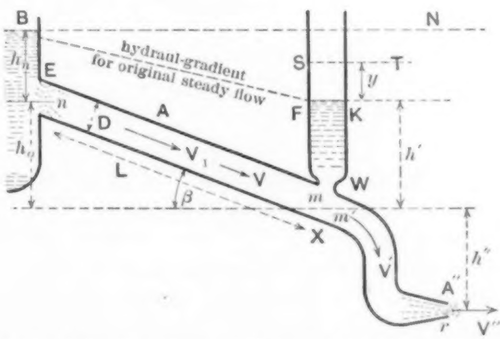

FIG. A SECTION OF CONDUIT AND STAND PIPE OF WATER POWER PLANT

ing and the water in the standpipe remains at rest, its level stands at *F K* and *B F* is the hydraulic gradient.

3 Let us now suppose that the "load" on the wheel is suddenly reduced below normal and is maintained constant at this lower value for an indefinite time; and that the governor and accessories are capable of maintaining a practically constant speed of wheel (equal to original) by varying the section of the jet, including a practically instantaneous change at the very outset. An unsteady flow now ensues in the main conduit, with gradual reduction of velocity *V* of the water in it, from the original value V_1, as the water rises in the standpipe. The velocity V' of the water at m', at the entrance of the short pipe $m'r$ (considered *to have the same sectional area at m'*,

viz: A, *as the long conduit*) after an initial sudden diminution from its original value V_1 to a new value V'_0, will thereafter as the water rises in the stand pipe continue to diminish, but the diminution is gradual. Finally, when the rise y in the stand pipe has reached its maximum value y_m, V and V' will have diminished to a common value which may be called V_2. The analysis now to be presented will have to do *only with this first upward movement of the water in the stand pipe*, in the attempt to determine the value of y_m and (incidentally) V_2. The following six equations refer to any instant during this unsteady flow, the variables being V, V', V'', y, and the pressures p_n and p_m at points n and m. Fig. A shows the meaning of the symbols h_0, h', h'', h_n, and β. Time is denoted by t (an element of time by dt).

4 Although the flow is now unsteady, position n, just inside the entrance of the main conduit, is so close to the réservoir that no sensible error will arise in using Bernoulli's Theorem for steady flow between surface B and position n; whence

$$\frac{p_a}{\gamma} + h_n = \frac{p_n}{\gamma} + (1 + \zeta_E)\frac{V^2}{2g} \qquad [50]$$

where ζ_E is the "coefficient of resistance" for the entrance of conduit p_a is atmospheric pressure, γ the weight of a cubic foot of water, and g the acceleration of gravity.

5 The inertia of the water in the long conduit EX is, of course, a very important element to bring into play. The net accelerating force acting on this cylinder of water, in the direction of its length, is $A\ p_n - A\ p_m + A\,l\,\gamma \sin\beta - \frac{f\,\pi\,D\,L\,\gamma\,V^2}{2g}$ where f is the "co-efficient of fluid friction." The mass accelerated is $AL\gamma \div g$ and the acceleration is $\frac{dV}{dt}$. Writing *accelerating force* = *mass* $\times$ *acceleration*, dividing through by $A\gamma$, and denoting $\frac{4f}{2g}\frac{L}{D}$ by C_0, we find

$$h_0 + \frac{p_n}{\gamma} - \frac{p_m}{\gamma} = \frac{L}{g}\cdot\frac{dV}{dt} + C_0 V^2 \qquad [51]$$

6 When the surface of the water in the standpipe or surge tank is passing a position $S\ T$, y feet above the original, $F\ K$, the pressure at the base m must overcome the weight of the column of height $h' + y$,

as also the atmospheric pressure p_a on the surface, and also accelerate ("overcome the inertia" of) the mass of that column with an acceleration $\frac{d^2y}{dt^2}$; and this leads to the relation

$$\frac{p_m}{\gamma} - \frac{p_a}{\gamma} = h' + y + \frac{h' + y}{g} \cdot \frac{d^2y}{dt^2} \qquad [52]$$

7 The pipe mr is supposed very short compared with the main conduit, and the flow from m' to r may hence be treated as a steady flow, justifying Bernoulli's Theorem; whence with ζ'' as the co-efficient of resistance for that pipe and nozzle, we have (neglecting V' as "velocity of approach")

$$\frac{p_m}{\gamma} - \frac{p_a}{\gamma} + h'' = (1 + \zeta'') \frac{V''^2}{2g} \qquad [53]$$

8 If Q cu. ft./sec. is the rate of flow through the nozzle at r, Q being also equal to $A''\ V'' = A V'$, we have, for the rising surge,

$$A\ V = Q + \frac{A}{R} \cdot \frac{dy}{dt} \qquad [54]$$

which may also be written

$$A\ V = A\ V' + \frac{A}{R} \cdot \frac{dy}{dt} \qquad [54a]$$

9 These are the fundamental relations needed for treating the problem, but it will be convenient to have also a relation holding good for the original steady flow, viz:

$$h_n + h_0 = h' + \left[\frac{1 + \zeta_E}{2g} + C_0\right] V_1^2 \qquad [55]$$

10 In the case of the long surge now under consideration the inertia of the water in the surge tank is of very slight consequence (as the writer has found by numerical trial) and hence in using equation [52] the term containing d^2y/dt^2 will be omitted.

11 To derive a relation between the three variables only y, V, and V', we combine equations [50] and [51] to get an expression for $\frac{p_m}{\gamma} - \frac{p_a}{\gamma}$

and then equate it to the right hand member of [52]; whence

$$h' + y = h_n + h_0 - \left[\frac{1 + \zeta_E}{2g} + C_0\right] V^2 - \frac{L}{g} \cdot \frac{dV}{dt} \qquad [56]$$

12 The coefficient (bracketed) of V^2 is practically the quantity C used by Mr. Johnson, and called by him the "friction factor," and will now be written as such. Combining [56] and [55] gives

$$y - C\,(V_1^2 - V^2) = -\frac{L}{g} \cdot \frac{dV}{dt} \qquad [57]$$

But, from [54a], we have

$$dy = R\,(V - V')\,dt \qquad [58]$$

and by multiplying [57] and [58] obtain

$$[y - C\,(V_1^2 - V^2)]\,dy = -\frac{RL}{g}\,(V - V')\,dV \qquad [59]$$

which, in general, is a differential equation between the three variables y, V, and V' (but V' is a function of y, due to the action of the governor, as will be seen).

13 During this first upward surge y varies from zero to the unknown y max. or y_m, and if we indicate the integration of the various terms in equation [59], just as it stands, denoting by V_2 the unknown common value of V and V' at the end of this surge, we obtain

$$\int_0^{y_m} y\,dy = -\frac{RL}{g}\left[\int_{V_1}^{V_2} V\,dV - \int_{V_1}^{V_2} V'dV\right] +$$

$$C\left[V_1^2 \int_0^{y_m} dy - \int_0^{y_m} V^2\,dy\right] \qquad [60]$$

14 From the point of view of exact mathematics equation [60] cannot be completely integrated unless V' is given as a function of V, and V as a function of y. Both of these functions are unknown. But since V' is a known function of y through the nature of the regulating apparatus, in this case of an impulse wheel, approximate methods are available, through equation [59], for finding y_m with sufficient accuracy for practical purposes.

15 Consider next the variation of V' as due to the action of the governor, in this specific case of an impulse wheel of the Pelton type. Let U = linear velocity of rim of the wheel (cup centers) at normal speed of wheel during the original steady flow, involving a definite value P (lb.) of the tangential "working force" acting on the cups. When the load on the wheel is diminished suddenly, the action of the governor will (instantaneously, say) so change the sectional area of the jet as to make the new value, P_0, of this working force smaller than P in the exact ratio of the reduction of load (otherwise the wheel would accelerate); and since the velocity V'' of the jet cannot increase until the surge begins to rise, this means that the rate of flow Q in the jet is suddenly made to take a value Q_0 cu. ft./sec. smaller than the original rate, and in proportion to the reduction of load.

16 From the mechanics of this impulse wheel, with e as its efficiency, we have just at the beginning of the surge, after the first quick action of the governor,

$$P_0 \text{ (lb.)} = \frac{2e\, Q_0\, \gamma}{g} (V_0'' - U) \qquad [61]$$

where V''_0 = velocity of the jet at this initial instant.

17 At any later instant, the "load" (at its reduced value) being supposed to remain constant, and hence P_0 also, we have, similarly,

$$P_0 = \frac{2e\, Q\gamma}{g} (V'' - U) \qquad [62]$$

containing two variables, Q and V''.

18 Equating [61] and [62],

$$Q = \frac{(V_0'' - U)\, Q'_0}{V'' - U};$$

or, since $Q = AV'$, and $Q_0 = AV_0'$,

$$V' = \frac{(V_0'' - U)\, V_0'}{V'' - U}; \qquad [63]$$

and this gives V' as function of V''. Also, from [52] and [53] we have V'' as function of y, viz:

$$V'' = \frac{1}{\sqrt{1 + \zeta''}} \cdot \sqrt{2\, g\, [h' + h'' + y]} \qquad [64]$$

19 Equations [63] and [64] and the differential equation [59] afford the means of plotting curves, with fairly close approximation, showing

the variation of V' with y, of V with y, and of V' with V; and thus of determining finally the value of y_m for any given set of numerical data; and a check on the result consists in the application of equation [60] after the curves mentioned have been drawn. The determination of these curves involves lengthy arithmetical and graphic operations.

20 Mr. Johnson treats equation [60] by assuming V' to remain constant (sufficiently so), presumably at the value assumed by it at the beginning of the surge, thus making $V' = V_2$; and by integrating the last term in an approximate manner, which perhaps may be expressed as follows: If the variable $C\ (V_1^2 - V^2)$ be denoted by Z, this last term may be written $\int_0^{y_m} Zdy$. When y is small Z is increasing slowly and dZ is less than dy; but later, as y nears its maximum, Z increases more rapidly than y, or $dZ > dy$, so that an approximate result may be reached by putting $dZ = dy$, as an average, throughout the whole summation; i. e.,

$$\int_0^{y_m} Zdy = \int_{Z_1}^{Z_2} ZdZ = \left.\frac{Z^2}{2}\right]_{Z_1}^{Z_2} =$$

$$\left.\frac{C^2}{2}(V_1^2 - V^2)^2\right]_{V_1}^{V_2} = \frac{C^2}{2}(V_1^2 - V_2^2)^2$$

And thus we are led to Mr. Johnson's equation [7], viz. (with present notation):

$$y_m^2 = \frac{RL}{g}(V_1 - V_2)^2 + C^2\ (V_1^2 - V_2^2)^2 \qquad [7]$$

in Par. 45 of his paper.

21 The writer will now give results that have been obtained by him in several numerical cases, beginning with those in which the design, while mechanically feasible, would be quite "un-practical" from an economical standpoint. It is thought, however, that the consideration of one or two cases of that character will lead to interesting and valuable mathematical indications. The units foot and second will be used.

22 *Case 1.* Let $L = 10\ 000$ ft. and $D = 4$ ft., with $h' + h'' = 80$ ft.; and velocity V_1 of original steady flow in main conduit $= 10$ ft./sec. The diameter of the surge tank is taken as 12 ft.; hence $R = 1/9$.

Let the conduit be of riveted steel in which the loss of head, at 10 ft./sec., is as high as 11 ft. per 1000 ft. of length (authentic in one instance). With these data we find $\frac{RL}{g} = 34.50$ and $C = 1.11$ That is, in the original steady flow the velocity is $V_1 = 10$ ft./sec. in the main conduit, the total head is 191 ft. and the total loss of head is 111 ft. Let the velocity at m' change suddenly from 10 to 8 by the action of the governor and let it be supposed in this present case (1) to remain constant at that figure. This makes $V_1 = 10$, $V_2 = 8$, and $V' = V_2 = 8$, constant.

23 The curve for V as obtained by the writer from these data is shown in Fig. 1, and gives by its intersection with the horizontal through the value 8 of the vertical scale $y_m = 39.7$ ft. By measuring

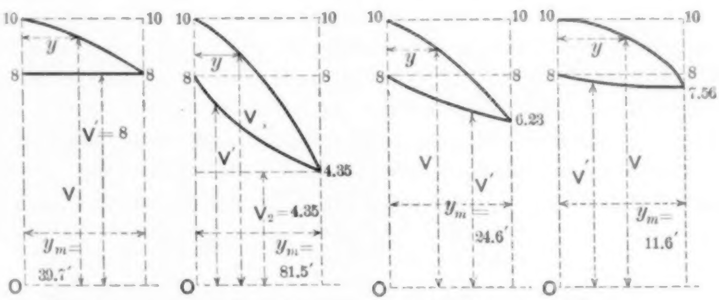

Fig. 1 Fig. 2 Fig. 3 Fig. 4

Diagram Illustrating Values of V

some ten values of V on this curve, properly distributed along the axis of abscissæ, it was found that the mean V^2 was 83.56. In equation [60] we now put $V' =$ constant $= 8$, $V_1 = 10$, and $\int_{V_1}^{V_2} V^2 dy = 83.56\, y_m$, etc.; that is,

$$\frac{y_m^2}{2} = -\ 34.50 \left[\frac{8^2 - 10^2}{2} - 8\ (8 - 10)\right] +$$

$$1.11 \left[100 y_m \ - \ 83.56\ y_m\right]$$

or, $y^2_m - 36.50\ y_m = 138$; whence, solving the quadratic, $y_m = 39.95$ ft., a fairly close check.

24 For this Case 1, Mr. Johnson's formula [7] gives 41.6 ft., which is seen to be only 4 per cent in excess of the other value derived by (fairly) rigorous methods.

25 *Case 2.* In this, and in the remaining cases, V' is to be variable, following the law of variation necessitated by the action of the governor in keeping the working force at rim of wheel constant at reduced load; as per equations [63] and [64].

26 Let the dimension values of L, D, and $h' + h''$, be as before, viz: 10 000 ft., 4 ft., and 80 ft., respectively; but for greater expedition in numerical work C is now taken a trifle smaller, viz: $= 1.00$, and $\frac{RL}{g} = 30$ (which means a slightly larger tank section). The tota head is now $80 + CV_1^2 = 180$ ft., and the loss of head $= 100$ ft. Take $V_1 = 10$ for original steady flow.

27 Now let the load on wheel be suddenly reduced by 20 per cent and hold the new value. The velocity V' will suddenly diminish from 10 to 8, the velocity of the jet being then, as during steady flow, of a value $V''_0 = 0.975 \sqrt{2g \times 80} = 70$ ft. / sec., and the (constant) velocity of the wheel cups one-half of this 70, or $U = 35$, which speed the governor will maintain. From equation [64] we have for any subsequent instant $V'' = 0.975 \sqrt{2g\,(80 + y)}$, while equation [63] gives

$$V' = \frac{(70 - 35) \times 8}{V'' - 35} = \frac{280}{V'' - 35}$$

28 A curve being plotted for V' and y as coördinates, and also (the differential equation [59] being then brought into play) a curve for V and y, the intersection of these two curves (see Fig. 2) gives y max. as 81.50 ft.; and a value of 4.35 ft./sec. for the final, common, value of V and V' at the end of the surge. This 4.35 is the V_2 of equation [60].

29 From the curve for V and y, 63.40 is obtained as the mean of all values of V^2 throughout the surge, with y as independent variable; while from the curve (now easily obtainable but not shown in Fig. 2) of V' and V the mean V' for V as independent variable is found to be 5.35. With these values placed in equation [60] and $V_1 = 10$, etc., there is obtained

$$\frac{y_m^2}{2} = 30\left[\frac{100 - 18.92}{2} + (5.35)\,(-5.65)\right] + 1.0\,[100\,y_m - 63.40\,y_m]$$

i. e., $y^2_m - 73.20\ y_m = 618.84$; and finally, $y_m = 80.85$ ft., as against 81.50 first obtained from the drawing.

30 If in applying formula [7] to this case we write $V_2 = 8$, the result is $y_m = 37.6$ ft. (54 per cent smaller than the 81.50); but if we write $V_2 = 4.35$ (although, as regards the use of [7], this value would be unknown in advance), $y_m = 86.7$ ft. is obtained, which is only 6 per cent in excess of the 81.50.

31 *Case 3.* We now take a somewhat more practical set of data, with $L = 5000$ ft., $D = 4$ ft., and a loss of head (for 48-in. riveted steel pipe) at the rate of 8 ft. per 1000 feet of length, with $V_1 = 10$ ft./sec., so that $C = 0.4$. Let the diameter of surge pipe be 16 ft. (nearly) leading to the value of 10 for $\dfrac{RL}{g}$. The whole head is 120 ft. up and the total loss of head is 40 ft. (which is still a large proportion, being one-third of total head); $h' + h'' = 80$ ft. and $V_1 = 10$.

32 As before, let the load suddenly diminish by 20 per cent, and remain constant at the new value so that V' suddenly changes from 10 to 8 ($= V'_0$). The two curves needed being drawn are found to intersect (see Fig. 3) at a point giving 24.60 ft. for y_m and 6.23 ft./sec. for the final V_2. The subsequent use of equation [60], with 74.52 as the mean V^2, and 6.81 as the mean V', gives a value of $y_m = 24.41$ ft., which is a close check on the above.

33 For the data of Case 3, if V_2 is considered as 8 in equation [7] the result of using [7] is $y_m = 15.7$ ft. (about 36 per cent too small); while if V_2 be taken as 6.23 (unknown beforehand, of course, in the use of equation [7]), a value of 27.2 ft. is found for y_m (about 11 per cent greater than the 24.6 above).

34 *Case 4.* In this case the data taken approximate to those of Mr. Johnson's numerical example in Par. 84. The total head is 225 ft., of which 25 ft. are lost in friction (here only one-ninth of total head), leaving $h' + h'' = 200$ ft. Length $L = 5000$ ft., $D = 6$ ft., that is, 72-inch riveted steel pipe, in which the loss of head is taken as 5 ft. per 1000 ft. of length when the velocity is 10. The diameter of surge tank is taken as 24 ft. (nearly) such that $\dfrac{RL}{g} = 10$; while with $V_1 = 10$ we have $CV^2_1 = 25$ ft.; i. e., $C = 0.25$. With a sudden 20 per cent reduction of load on the wheel, we have $V'_0 = 8$ ft./sec.; the load to remain constant at the reduced amount. The curves found for V' and V are shown in Fig. 4. The V curve is seen to be more "rounding" than in the other cases (and it should be said that in all these cases this curve must have a vertical tangent at the right-

hand extremity, since at that point the value of $dV \div dy$ from equation [59] must be infinite, $V - V'$ being = zero). V_2, the final value common to V and V', is found to be 7.56; and y_m = 11.55 ft. With the value of 86.10 for the mean V^2 and 7.675 for the mean V', obtained from the proper curves, equation [60], after solution of the quadratic, gives y_m = 11.6 ft.

35 On substitution in equation [7] we find that if V_2 is considered to be 8, y_m = 11.0 ft., which is only 5 per cent less than 11.6; while if V_2 is taken to be 7.56 (the actual final value of both V and V_1) we obtain y_m = 13.2 ft. (some 14 per cent in excess of the 11.6).

36 On reviewing Cases 2, 3, and 4, it is seen that in using equation [7] if we regard the change of velocity occurring in the main conduit during this first upward surge to be in proportion to the reduction of load on the wheel (i. e., putting $V_2 = 8$ each time) the margin of error is small in 4, which is the most practical case; and very considerable in the other two, especially in Case 2, which is, however, a decidedly "un-practical" case. On the other hand, if for V_2 in [7] we substitute the actual final value of V in main conduit, only 6 per cent error is found in Case 2, 11 per cent in 3 (more practical), and 14 per cent in 4 (most practical). It need hardly be said, perhaps, that equation [7] was not intended to apply to such data (with relatively large friction-heads) as are found in Cases 1 and 2.

37 The writer has supposed a single Pelton impulse wheel to be in use in each of these numerical illustrations; not because that type of wheel might be considered advisable for heads of 80 to 200 ft., but because the data and treatment could thereby be made much more specific, as regards the action of the governor, than in the case of a turbine. Of course, with such a type of motor several wheels of the same radius and on the same shaft, or a few wheels with many nozzles, would be necessary to take care of the water in the foregoing instances. For simplicity the writer has considered only one wheel and one nozzle, but the result is the same.

38 Perhaps the foregoing numerical results may serve as rough guides as to what extreme change of velocity in the main conduit might advisably be assumed in using equation [7] in connection with surge tanks.

THE AUTHOR The purpose of my little monograph on Surge Tanks was to awaken some much-needed enthusiasm on the subject, and to call forth discussion and adverse criticism, in the belief that the resulting matter when coördinated would not only add to my own

store of information, but present a collection of ideas sufficiently varied to enable any interested student, to separate the wheat from the chaff and arrive at a correct understanding of these complicated phenomena. The outcome thus far is highly gratifying, but still a long way from the result sought. Mr. Larner's work is of great practical value if properly interpreted.

2 My paper was prepared at very short notice and is decidedly incomplete when considered in the light of a text-book, and deficient in what might be instructive examples as to the limitations of its equations. After perusal of the work of my critics, I want to say at the outset, however, that I find absolutely no reason for altering any of my equations, unless it were to make them universally exact, a difficult task which has not been done in the discussion, as I shall proceed to show. Equations 7 and 12 called forth the most comment, and therefore I shall devote myself to a short general discussion of these two equations in the light of criticism. They are just what they purport to be, nothing more nor less, and their accuracy according to the preliminary assumption is unquestionable. Both of them assume a constant draft velocity as the superior limit of the integration, and hence have only an indirect value when applied to an actual case where the draft velocity varies (particularly in the simple tank regulation) according to the action of the governor in its effort to maintain constant power. I described this effect at great length, and yet the language of my critics in some passages would convey the impression to any but a very careful reader, that I had overlooked this point and attributed to my equations a perfection entirely unwarranted by facts.

3 It is gratifying to note the close agreement between Mr. Harza's formula and the preliminary ones in my paper, and it is to be hoped that his work of analyzing and checking will be continued to include the differential principle, and the use of compressed air.

4 I have not given the attention to the simple stand pipe that Mr. Harza has, because I have always regarded the mathematics relating to it as simply a step toward the discovery of a regulator which would meet more perfectly the requirements of practice. Such a regulator, I believe, is to be found through a study of the "differential principle," and I have yet to be convinced that the regulator I propose is not far better than the simple stand pipe, despite Mr. Harza's argument somewhat to the contrary. His reasoning, in the abstract, I believe is sound, but it shows a possibility of modification after more detailed study of the question with the essential aid of

definite dimensions, which may be found by tedious processes of arithmetical integration. The *tendency* to produce worse regulation conditions by the introduction of a resistance between surge tank and conduit certainly exists, and is due to conditions very much as stated by Mr. Harza. An appreciably bad effect upon regulation never gets to the point of appearing, however, because lost sight of when compared with the manifest advantages. Mr. Harza shows a realization of the possibility of this condition in Par. 47. I have no doubt that further study of the question on his part will largely clear up this slight difference of opinion.

5 I have never hit upon a method of approximation which I regarded as accurate enough to publish, such as Mr. Harza's equation (20), and was compelled to leave the effect of the governor's action to the judgment of the designer in selecting his value of $V_2 - V_1$. Another reason for not giving much attention to the governor action is that it does not enter into the problem in the same way when the differential principle is used. In this case, it is necessary only to select a value for V_2 in excess of that which one *expects* to represent the maximum load change for a constant head, as pointed out in Par. 55.

6 As to the argument (Par. 36) against catching all the energized water, I should like to point out two things: First, it does not appear clear to me that curtailing the upward surge, whose maximum occurs for rejected loads, puts any limit on the downward surge, whose maximum occurs for demanded loads, except where a super-added or a perpetual wave exists. The latter condition is possible where the differential action is omitted, and where one is really designing for minimum size of tank consistent with regulation requirements. In that case, water would continue to spill at regularly recurring intervals all day, or until another load change happened to occur in such a way as to fit into the existing wave and smother it. Mr. Harza had in mind the "super-added" wave, but even this is far more likely to occur when the differential action is omitted. It was probably not intended to imply the possibility of such an obviously poor design when arguing its advantage.

7 Second, the saving of water at the rare intervals of "full load rejected" is not of much consequence, to be sure; but the saving of the cost of construction work necessary to catch and lead away a possible large discharge due to a shut-down is of such importance as to weigh decidedly in the balance against the additional cost of surge tank to catch and hold all the energized water. Allowing the water

to spill over has a damping effect upon surge waves for rejected loads, as he states, but this can be accomplished much more effectually by means of the differential principle without waste of water and usually at a saving in cost besides. It is possible, through such a regulator, if designed especially for maximum damping effect, to make the surge wave strictly and absolutely dead beat; and the importance of a thorough study of this principle cannot be overestimated.

8 Referring now to Mr. Harza's Par. 49, I want to say in justice to Mr. Warren's valuable paper that my remarks were not intended to attain the dignity of a criticism, but merely as introductory to a line of argument which seemed to be new to him. I am still of the opinion that a properly designed regulator, or system of regulators, is actually "a complete remedy for the troubles in speed regulation caused by the excessive inertia of a water column." Mr. Harza means, I judge, that the surge tank cannot be made a complete remedy for variation in head—but it is obvious that a moderate amount of head variation does not at all mean "troubles in speed regulation;" for otherwise, where do his own equations lead? His criticism of my work is very creditable and fair, it seems to me. The values obtained by my Equation 7 are not as close to the truth, in the instances cited, as those obtained by his Equation 20, as might be expected; although, as I shall show later, Equation 7 gives closer results when applied to Professor Church's examples.

9 These problems contain an inordinate amount of friction, which is rarely more than 10 per cent for a correctly designed plant, and often as low as 5 per cent.

10 I shall adopt Professor Church's nomenclature for this little study, with the addition of V_3 which he does not use but which I need as a symbol for the steady velocity under the new load after all vibrations have ceased. Let H be the whole head; then from the principle of constant power we have, neglecting the change in wheel efficiency

$$(H - CV_1^2)\, V_0^1 = V_3 H - CV_3^3 \qquad (23)$$

also $C\,(V_1^2 - V_3^2)$ = the net change in gradient or the difference in head before and after the surge, which one may call Δh.

11 Now a very little thought at this juncture will disclose the fact that y_{max} must be at least as great, in all cases, as Δh; it is also nearly as plain, I think, that y_{max} is always greater than Δh; however, for my present purpose, I am content to stop with the former statement which admits of no argument, for I propose to solve for V_3 in each of Professor Church's three cases, and to show that Δh is

greater than his y_{max} in all except the last case. This, of course, leads to but one conclusion, which is that his values are too small. I do not doubt, however, that they are as accurate as one could figure them

TABLE 1

	V_3	Δh	V_2	y_{max}
Case 2	3.88	84.95	4.35	81.50
Case 3	6.10	25.00	6.23	24.60
Case 4	7.60	10.60	7.56	11.60

by the methods adopted by him, and they are undoubtedly near the truth, but not quite so near in two cases as could have been obtained directly by this equation (23), just written. Of course, this equation is of no value whatever in determining y_{max} except where $\frac{c}{R}$ is so very large that y_{max} is not sensibly larger than Δh.

12 By trial in Equation 23, the following values are found for V_3 and Δh for the three cases and I have also listed for comparison Professor Church's values for his V_2 and y_{max}.

Example (case 2):

$$H - CV_1^2 = 80$$
$$V'_0 = 8$$
$$H = 180$$
$$C = 1.00$$
$$640 = 180\ V_3 - V_3^3, \text{ from which by trial } V_3 = 3.88$$
$$\Delta h = (100 - 15.05) = 84.95$$

13 Let us now list the values for y_{max} as obtained from my Equation 7, putting V_3 (*known before hand*) as the superior limit, and compare them with the values for y_{max} obtained from Mr. Harza's Equation 20, putting his $V_2 = V_3$ above. He states in Par. 23 that the upward surge is always less than the value given by his Equation 20. Now these are examples of upward surge, and yet it is seen that Equation 20 gives not only smaller values than the true y_{max} but even smaller than Δh, which is itself necessarily at least a little less than the true y_{max} in all cases. It is also seen that Equation 7 gives far closer results in these instances than the more exact equation of Mr. Harza's. This would not ordinarily be true, however, in a practical example.

14 The correct values for y_{max} probably lie between Columns 2 and 3. These computations are all made on an 8 in. slide rule, but are dresumably otherwise accurate. The comparison is not made for

TABLE 2

	Equation 20 $y_{max.}$	Equation 7 $y_{max.}$	h	Professor Church y_{max}
Case 2	46.60	91.4	85.0	81.5
Case 3	16.00	27.9	25.0	24.6
Case 4	8.75	13.0	10.6	11.6

the purpose of discrediting Mr. Harza's Equation 20, which is a valuable one when judiciously used, just as my own are.

15 One always knows beforehand the values of V'_{max} and V_3 (since y_{max} is selected to be the result of the design). By substituting each in turn if necessary for V_2, one can usually, in a *practical case*, fix the true value of the surge within limits, and thus be sure of being at least safer than by the use of any formula that purports to include all the variable quantities and only partially does so. It should not usually be found necessary to make any material alteration in tank dimensions after checking up by arithmetic integration, as I have repeatedly proved. Therefore, for my own use, I prefer a simple equation like 7, which has the advantage of being fairly accurate for what it purports to show, to a more exact one which only partially does what it claims, always leaving one in some doubt as to the limits between which the true result lies. The following figures will assist in making a little more careful study of the numerical example of Par. 87, in order to help demonstrate the usefulness of Equation 7, when applied with discretion.

16 First, let us solve for V'_0 and V_3; we have

$$V_1 = 13$$

$$V'_{max} = 15$$

$$h = \text{working head for } V_1 \text{ or } V'_0$$

$$H = h + C\,V_1^2 = 216.9$$

$$h \text{ for } V'_{max} = 190$$

$$190 \times 15 = 216.9\,V_3 - \tfrac{1}{10}\,V_3^3$$

$$V_3 = 14.55$$

$$190 \times 15 = V'_0 \times 200$$

$$V'_0 = 14.25$$

$$\Delta h = \tfrac{1}{10}\,(\overline{14.55}^2 - \overline{13}^2) = 4.25 < y_{max}$$

Equation 7, with $V_2 = V_3$, gives y_{max} = about 7.75
Mr. Harza's 20, with $V_2 = V_3$, gives y_{max} = about 8.25
Equation 7, with $V_2 = V'_{max}$, gives y_{max} ± about 10.00

17 None of the values are right, but Mr. Harza's value is without doubt the nearest, though presumably small. If the correct value is about 9, then it is apparent that a tank somewhat smaller than 34 ft. could be used, and therefore my numerical comparison between the simple tank and the differential is rather unfavorable to the former; just how much I cannot say without more work than is possible within the time at present available. Pressure of time made it necessary in the first instance to close my paper almost before I had started any numerical illustrations, and thinking these power computations for different heads elementary, I contented myself with a hurried illustration, believing that any one who went into the subject sufficiently to understand it would have no difficulty in getting some good out of my formulae; for the others, probably no amount of explanation would have sufficed, and the explanation might have been fully as misleading to them as the work I did.

18 The value of R is very sensitive to the magnitude of V_2 and an absolute computation for it is perhaps impossible by any reasonable process; but y_{max} is not so sensitive, and y_{max} is really what one wants to foretell within reasonable limits. It must be remembered also that V_2 is only a guess in the first place, supposedly of sufficient size as compared with V_1 to represent a maximum load change. My study of the vagaries of Equation 7 have not been extensive enough to warrant my writing at any great length. My belief in the superiority of the differential scheme has led me to work with equation 12 for the most part, much to the neglect of Equation 7. I believe, however, that it is a valuable equation for use in connection with Mr. Harza's Equation 20, and probably a combination of the two can be worked out which would be quite accurate for all practical cases.

19 An important point in favor of the differential form of regulator, which I inadvertently omitted to mention, is that since the maximum surge is less than in the simple form a smaller water wheel is required to pick up full load as the head recedes; this works two ways to advantage: the units are less expensive in the first place, and the wheels operate continuously at larger gateage and hence at better normal or average efficiency.

20 The discussions indicate that the equations of the paper can-

not be satisfactorily used except by those who have given the subject a great deal of study, and attention should be called to one or two vital facts. The symbol V_2 in the foregoing formulæ denotes a hypothetical velocity which does not exist in practice, although it is very easily conceivable from a mathematical standpoint. The equations from necessity are worked out upon the assumption that while the wheel is operating steadily under a conduit velocity of V_1, a sudden demand for more load causes the draft velocity in the penstock near the wheel to change instantly to V_2 and remain at that value indefinitely, or until steady flow at this new rate is established. The action of the governor and the variation in pressure head both prevent a realization of this mathematical ideal, but the results obtained in this way are, nevertheless, very satisfactory if one makes a proper assumption, governed by his judgment for the value of V_2.

21 If, for example, one substitutes for V_2 the maximum value of the draft velocity under the depleted head, at the end of the first quarter cycle of the pressure wave (as in Par. 55), the resulting values derived from Equation 12 will be very accurate. The same treatment with Equation 7 will give values always on the safe side, but *in a practical case*, not far from the truth. In other words, the tank capacity thus figured will take care of a little larger load change than contemplated. It is certainly useless to attempt further refinement of Equation 7 without taking into consideration the length of life of the pressure wave, for, in many seemingly practical examples, this feature may completely vitiate the efficacy of the simple surge tank, and, even with the factor of safety thus somewhat inadvertently involved, one is still uncertain of the outcome, unless recourse is had to other considerations. It is the opinion of the writer that the differential principle ought seldom if ever to be omitted, in which case Equation 7 is not needed except for scouting purposes. None of the equations are recommended for handbook use.

Additional discussions upon this paper were contributed by Mr. Chester W. Larner, Mr. Morris Knowles and the author and published in The Journal for October, 1908, January and June, 1909.—EDITOR.

MONTHLY MEETINGS

HELD IN NEW YORK
OCTOBER 13 AND NOVEMBER 10, 1908

REGULAR MONTHLY MEETINGS

THE OCTOBER MEETING

The October meeting of the Society was in charge of the Gas Power Section. It was held in the Engineering Societies Building on Tuesday evening, October 13, Dr. C. E. Lucke, Chairman of the Section, presiding. A report for the Membership Committee was given by Mr. George A. Orrok, showing an increase of 123 in the membership of the Section since September 1.

The opening discussion was upon the communication from Mr. H. L. Doherty, Chairman of the Meetings Committee of the Section, published in the October number of the Journal, outlining a definite plan of action for the Section. Messrs. Tait, Rushmore, Bump, Wilkinson, Dr. Lucke and Professor Reeve participated in the discussion. Suggestions were made particularly urging that information upon the reliability of gas engine installations be authentic: to the effect that engineering data could be effectively distributed bp the adoption of a "Question Box;" that data should be gathered and filed at Society headquarters; and that questions to be considered by the Section should include those involving precise knowledge and investigation as well as constructive and operative problems.

The Progress Report of the Committee on Standards, published in the September Journal, was discussed at length by Messrs. Lummis and Bibbins, and Professor Reeve, the discussion hinging mainly on gas engine efficiency and heat value of gas, with arguments for the use of total heat values and effective heat values.

The first paper of the evening was by E. A. Harvey on Bituminous Producer Plants, and gave data upon the cost and performance of three different plants. This was illustrated by lantern slides and discussed by Messrs. Tait, Parker and Bibbins. A paper, The Loss in Fuel Weight in a Freshly Charged Producer, by N. T. Harrington, owing to the lateness of the hour, was not discussed.

THE NOVEMB .R MEETING

On the evening of November 10 a paper was presented by Mr. Franklin Phillips of Newark, N. J., upon The High Powered Rifle

and its Ammunition. Lantern slides showed details of the mechanism of modern types of rifles and ammunition, the results of target practice, and methods followed by marksmen in shooting.

Mr. Fred J. Miller, Vice-President, presided. In introducing the speaker he said that as an engineer and a manufacturer he needed no introduction, but that many might not know that he was an expert in rifle practice and an instructor in the art in the New Jersey National Guard.

The High-Powered Rifle and its Ammunition

The speaker stated that the present high-powered rifle is the outcome of the invention of the jacketed bullet by Major Rubin of Switzerland in 1883–1886. About 1890 the Krag-Jorgenson rifle was selected by the Ordnance Board for the United States Army, which marked the beginning of the use of high-powered rifles for military purposes in this country.

This rifle, as well as all of the more modern rifles, is known as the bolt gun. The breech action is similar in its movements to that of the common door bolt. The bolt is pushed forward in a line parallel with the axis of the bore and closed by giving the knob at one side of the bolt an angular movement to lock it in position. In arms of this kind the act of unlocking the breech block cocks the firing pin. The barrel of the Krag rifle is 30-caliber in its smallest diameter. It has four grooves, each 0.004 in. deep, and the width of the land is $\frac{1}{3}$ that of the groove. The diameter of the bore is 0.308 in.

With the introduction of a higher powered rifle came possibilities for a revival of long range shooting, which had been a lost art in this country since the time of the famous matches at Creedmoor, and the New Jersey Rifle Association, composed of members, active and retired, of the National Guard of several states, challenged the Ulster Rifle Association of Ireland to a long range match for the historic Palma trophy.

This match was stated by the speaker to mark the beginning of the improvement of the military arm and its ammunition in the United States. The Americans shot the match with rifles made for them by the Remington Arms Co., and the Irish team used the Roumanian Mannlicher arm with Austrian ammunition. At the same time the National Rifle Association of America issued a challenge for the Palma trophy to the military nations of the world, which was accepted by a team from Canada.

Both the visiting teams won handsomely over their American opponents.

After the matches the Irish team allowed two American experts, Mr. William Hayes and Dr. Walter G. Hudson, to examine their Mannlicher rifles and ammunition. The bore of the barrel was calibrated by pushing through it a lead bullet of slightly larger diameter than the bore, after which the diameter of the bullet was measured.

The ammunition was also calibrated and found to be almost 0.001 in. larger than the caliber of the gun so that the bullet formed a gas-tight piston in its passage through the barrel. It was further found that the gun bands which secured the barrel to the stock contained a lining of chamois which was sufficiently elastic to allow the barrel to expand when heated from firing, without becoming distorted.

It was found that the barrels of the Krag rifles were cramped by having the bands too tight and that the ammunition used was smaller than the normal caliber of the arm. This caused the bullet to act like a leaky piston and the powder gases under the enormous pressure of 36 000 lb. per sq. in. would blow past the bullet, causing irregular results in shooting and tending to foul the barrel with a deposit.

It was further found that in long range shooting the bullet upset in its flight, making an aperture in the target suggestive of a key-hole, which gave the name "key-hole shot" when the bullet was fired under such conditions. There were three ways of overcoming this. One was to increase the powder charge in the gun; another to decrease the length of the bullet, making it lighter; and the third to change the twist of the rifling. It was found that an increase of the powder charge by about $2\frac{1}{4}$ to $2\frac{1}{2}$ grains, and the use of a 220-grain bullet, produced the best results. The diameter of the bullet was also increased to 0.3085.

After these changes had been made a return match for the Palma trophy was shot in Canada with competing teams from several countries. While Great Britain won the match, the American team did very much better than before and the performance of their arms was most creditable.

Many years' experience has led ballistic experts to believe that the danger zone of an arm would be materially increased by flattening the trajectory, and the invention of the pointed bullet now used was a marked advance in this direction. The 1903 model of the Springfield rifle and the 1907 ammunition are late developments in this direction. A feature of the barrel is its shortness, so that the gun can be used by all branches of service, including the cavalry. A

muzzle velocity of 2700 ft. per sec. is obtained, and to show the power of penetration the speaker mentioned that in a skirmish by the National Guard at Somerville, N. J., a bullet fired at a 200-yd. range passed entirely through the lower flange of a railroad rail which formed the coping of the railroad pit. The steel was at least ⅜ in. thick where the bullet went through, making a clean, round hole without any ragged edges. This type of rifle is a modification of the Mannlicher, which has some points of superiority over the Krag. The speaker said this rifle was the best constructed piece of gun mechanism ever turned out in quantity by any armory. It has a remarkable record, notably in the longer ranges, very much less elevation being required on account of the quicker flight of the bullet and its pointed shape. Serious difficulties have been experienced, however, from the fouling of the barrel occasioned by the high temperature of the powder gases.

Mr. Phillips traced the development of gun sights. The 1884 model of 45-caliber Springfield rifle had a fine adjustable sight, but when the Krag rifle was adopted it was issued with the plain bar sight, having a V notch, and without any windage gage. This was displaced by the Phipps sight and in the 1903 model by the Dixon sight; and this in turn by one also known as the Phipps sight, which has adjustments enabling much better marksmanship. The speaker described a sight of his own invention in which the zero base is made movable and adjustable so that the arm will hit an objective point when on its zero reading.

DISCUSSION

The paper was discussed by Capt. Kellogg K. V. Casey of the E. I. Du Pont de Nemours Powder Co., who presented some data upon ammunition. He said that the utility of the 150-grain bullet used with the new rifle had been doubted, owing to the effect of the wind upon its flight, although the tables of the Ordnance Department and of rifle manufacturers were to the effect that a bullet of this weight would make the best flight. Efforts have been made to determine whether a heavier bullet would be more effective and he had found that one which weighed 180 grains would be the largest that could be used without projecting into the powder chamber. Bullets of this weight would have started at a velocity less than 2700 ft. per sec., otherwise there might be too great an increase in the pressure in the rifle. Formerly, when a barrel 30 in. long was used, 43 000 lb. per sq.

in. was considered the highest allowable pressure; but was increased to 46 000 lb. per sq. in. when the shorter barrel was introduced, on account of the loss in velocity due to the shorter barrel. To secure a velocity of 100 ft. with a 180-grain bullet would require a pressure of 52 000 lb. per sq. in.

The problem was to get a charge that would give such a velocity that with the 180-grain bullet there would be the same angle of departure with a given time of flight as with the 150-grain bullet starting with a velocity of 2700 ft. In a series of tests conducted in New Jersey it was found that a velocity of about 2500 ft. would give the same angle of departure and time of flight in a thousand yards as 2700 ft. with the 150-grain bullet. The performance with a heavier bullet under these conditions was better and the fire more accurate than with a heavier bullet. The future use of a heavier bullet was predicted. The speaker did not attempt to explain the apparently contradictory results obtained with the heavier bullet under a low initial velocity.

The paper was also discussed by Mr. Frederick A. Waldron, who stated that he was retained by the Ross Rifle Co., of Quebec, Canada, which manufactures the rifles for the Canadian government, and he gave some data of interest regarding this arm. It has a muzzle velocity of 3100 ft. per sec. and uses powder of the guncotton variety. The bullet is 0.28 in. in diameter and weighs from 140 to 160 grains. The trajectory at 800 yd. is 5 ft. 4 in., and the temperature of combustion with the powder used is so low that there is no trouble from erosion.

No. 1205

LOSS OF FUEL WEIGHT IN A FRESHLY CHARGED PRODUCER

By N. T. Harrington, Lansing, Mich.
Member of the Society

The object of the test was to determine the relation between the coal fired and that actually consumed during the first few hours of operation, after completely filling the producer with a new charge of fresh coal. The apparatus used was a suction gas producer, and gas engine, arranged as shown in sectional drawing in Fig. 1.

2 The generator shell was mounted on platform scales, that the loss in weight due to combustion might be read directly. Its gas delivery pipe was connected to the scrubber by means of a water seal to allow free movement of the generator with the platform of the scales. All connections were flexible. A prony brake was used on the engine and the load was kept as constant as possible.

3 The coal was analyzed from the pile, and from samples taken as fired during the actual test. The ash as drawn from the producer, and the salvage as returned to the producer, were also sampled and analyzed. The gas was analyzed every hour. The final charge withdrawn from the producer at the end of the run was analyzed to show the increase in the amount of ash.

4 The test was started by weighing the producer empty and clean. The fire was then started and the producer filled with fresh coal, after which its weight was again taken. After blowing hot with compressed air to obtain good gas, the fire was poked enough to ensure a compact fuel bed, and the producer again filled. The engine was then started and the weight of the generator taken simultaneously.

5 Throughout the run all fresh coal added to the producer and the ash and salvage taken from the producers were weighed on separate scales. The fire was cleaned and the producer refilled every

Presented before the Gas Power Section at the Monthly Meeting (October 13, 1908), of The American Society of Mechanical Engineers.

3½ hours. The loss in weight of the fuel in the producer and the brake load on the engine were noted every 15 minutes. All weigh-

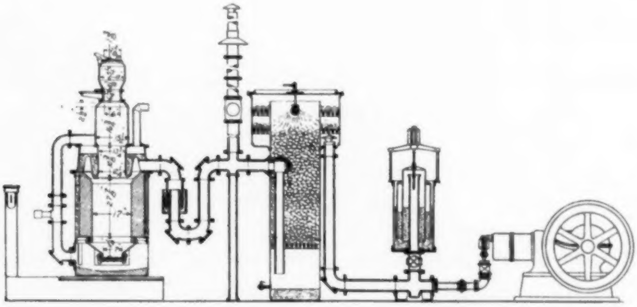

Fig. 1 Sectional Drawing of Suction Gas Producer and Gas Engine Shown Below

ings of producer and charge were so made and calculated that the consumption of combustible could be recorded every 15 minutes,

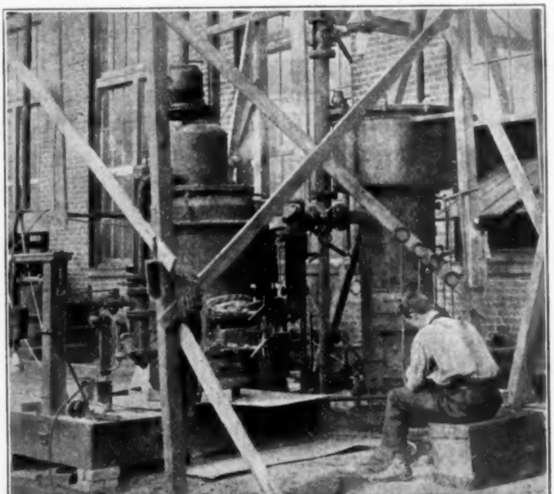

Fig. 2 Suction Gas Producer and Gas Engine

correction being made for the ash and salvage drawn from the producer, and the salvage returned to the producer.

6 The water in the vaporizer, and the water in the seal between the generator and scrubber were kept at constant levels.

7 At the end of the run the fire was cleaned and the producer filled, after which engine and producer were shut down and the final weight of producer and charge taken immediately. The entire charge

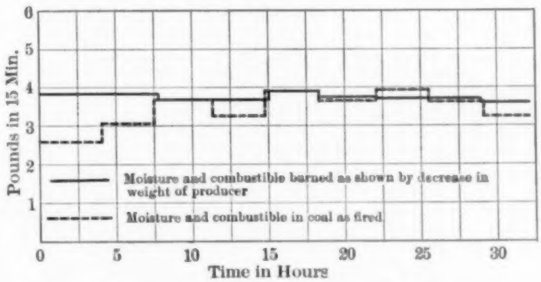

FIG 3 RECORD OF COMBUSTIBLE AND MOISTURE

with the exception of the clinkers sticking to the walls was then withdrawn from the producer and quenched, and the producer again weighed to determine the amount of clinker sticking to the walls. The clinkers were then removed, pulverized, and mixed with the rest of the charge and the whole sampled and analyzed. By this means

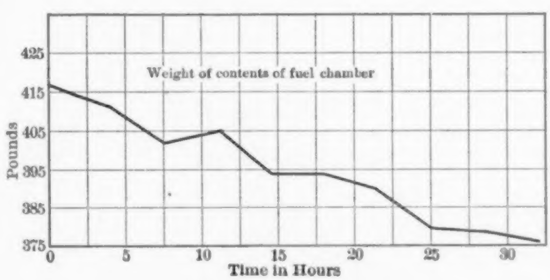

FIG. 4 RECORD OF LOSS OF WEIGHT OF CHARGE

it was possible to strike a balance between what was put into the producer and what was taken out.

8 Fig. 3 shows the relation between the combustible burned during the nine periods of the test, corresponding to the nine times the fire was cleaned and producer filled, and the combustible fired to the producers during the same periods. Figures for combustible were taken to include moisture. The combustible burned was deter-

mined by the loss in weight shown by the scales under the generator. The combustible fired was determined by deducting the ash contents from the weight of coal as fired, and weighed separately on small scales.

9 While these two curves do not show the actual amounts of coal fired and consumed, it may be assumed that the ordinates are directly proportional.

10 Fig. 4 shows the actual loss in weight of the charge in the producer by periods.

11 Care was taken that the fire was clean and without cavities at the times of filling, and that no arches or stoppages occurred in the fire at any time during the test. At the end of the run the fire was poked vigorously from the top to determine how much additional coal could be jammed into the producer. Only 10 lb. more could be added, and it is possible that this slight settling of the coal was due to a portion of the charge being forced up into the gas ring.

12 The difference of 40 lb. between the weight of the charge at the beginning and end of the run is to be attributed to the burning out of the carbon and volatiles from the lumps of coal, leaving the laminated ash structure to occupy about the same space as the original lumps of coal but increasing the ash contents of the mass.

13 For a short run with a fresh fire, it would seem that the coal fired to the producer does not represent the actual coal consumed and that in a 30-hr. run it may be as much as 8 per cent too low.

General Data

PRODUCER

Duration of test	32 hr.
Diameter of fuel body	16 in.
Area of section of fuel body	1.395 sq. ft.
Weight of generator empty	3491 lb.

ENGINE

Single cylinder, single acting, four cycle

Diameter cylinder	$7\frac{7}{8}$ in.
Stroke	$10\frac{1}{8}$ in.
R.p.m.	380
Average b.h.p. developed	12.6

COAL ANALYSIS

	Percentage coal as fired	Percentage final charge drawn from producer	Percentage ash
Moisture	2.58	.....	.10
Volatile carbon	5.47	2.72	1.68
Fixed carbon	74.09	75.49	28.42
Ash	17.86	21.79	69.80

RESULTS

Weight of charge in producer at the time of starting run, pounds	416	Loss of combustible and moisture by weight during run, pounds	471.15
Weight of coal added during run, pounds	506	Weight of ash withdrawn during run, pounds	76.75
Total	922	Weight of final charge withdrawn from generator, pounds	376.00
Error	1.9		
Total, pounds	923.9	Total, pounds	923.90

Weight of charge in generator at beginning of run, pounds	416
Weight of final charge withdrawn from producer, pounds	376
Loss in weight of charge due to fire loading up with ash, pounds	40

No. 1206

THE ANNUAL MEETING

PROGRAM

OPENING SESSION

Tuesday, December 1, 7.45 p.m., Auditorium

THE PRESIDENT'S ADDRESS

THE CONSERVATION IDEA AS APPLIED TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, M. L. Holman.

CONFERRING HONORARY MEMBERSHIP

Honorary membership in the Society was conferred upon Dr. John A. Brashear, after which Dr. Brashear made an address on the Photography of the Stars, illustrated by beautiful lantern slides.

PROFESSIONAL SESSION

Wednesday, December 2, 9.30 a.m., Auditorium

Annual business meeting. Reports of the Council, Tellers, and Standing and Special Committees.

PAPERS

THE ENGINEER AND THE PEOPLE, Morris Llewellyn Cooke.

Discussed by,

A. C. Humphreys, Talcott Williams, Geo. W. Guthrie, F. W. Taylor, Arthur T. Hadley, F. M. Day, H. F. J. Porter, A. L. Church, A. W. Moseley, J. M. Dodge, Ambrose Swasey, C. W. Hunt, F. R. Hutton, Oberlin Smith.

THE PRESENT STATUS OF MILITARY AËRONAUTICS, Major Geo. O. Squier, Acting Chief Signal Officer, U. S. A.

Discussed by,

W. J. Humphreys, J. A. Brashear, G. L. Fowler.

This was the first presentation of the subject of aëronautics before a national engineering society in America. By reason of his con-

nection with the Signal Service, Major Squier has had an opportunity to observe at close range the construction, equipment and principles of operation of heavier-than-air machines and dirigible balloons. The material upon aviation on file at the War Department, including the data sent in by the attachés in different countries, is the most complete in this country, and was at the disposal of Major Squier for the preparation of his paper.

Luncheon was served to members and guests.

PROFESSIONAL SESSION

Wednesday, December 2, 2 p.m., Auditorium

STEAM AND POWER PLANT PAPERS

A METHOD OF OBTAINING RATIOS OF SPECIFIC HEAT OF VAPORS, A. R. Dodge.

Discussed by,
H. N. Davis, S. A. Moss.

THE TOTAL HEAT OF SATURATED STEAM, Dr. Harvey N. Davis.

Discussed by,
C. H. Peabody, Wm. D. Ennis, R. C. H. Heck, L. S. Marks, I. N. Hollis, C. C. Thomas.

FUEL ECONOMY TESTS AT A LARGE OIL BURNING ELECTRIC PLANT, C. R. Weymouth.

UNNECESSARY LOSSES IN FIRING FUEL OIL, C. R. Weymouth.

Discussed by,
Geo. H. Barrus, Wm. Kent, Wm. D. Ennis, J. R. Bibbins, I. E. Moultrop, A. H. Kruesi, F. W. O'Neil.

LECTURE

Wednesday, 8.15 p.m., Auditorium

AËRONAUTICS, Lieut. Frank P. Lahm, of the Signal Corps, U. S. A., a member of the Aëronautical Board.

This was an illustrated lecture of interest to every member as well as to the ladies. Lieutenant Lahm has for many years experimented with dirigible balloons for war purposes and has participated in several international balloon races, winning the James Gordon Bennett cup in 1906. He has taken part in the experiments at St. Louis, and has made ascensions with Orville Wright at Fort Myer, Va.

PROFESSIONAL SESSION

Thursday, December 3, 9.30 a.m., Auditorium

MACHINE SHOP PRACTICE

Efficiency Tests of Milling Machines and Milling Cutters, A. L. DeLeeuw.

Discussed by,

F. J. Miller, Wilfred Lewis and W. H. Taylor, F. W. Taylor, H. W. Hibbard, J. J. Flather.

The Development of a High Speed Milling Cutter, Wilfred Lewis and W. H. Taylor.

Discussed by,

Fred J. Miller, Oberlin Smith, F. W. Taylor, A. L. DeLeeuw, A. B. Carhart, R. T. Stewart, W. J. Burkitt.

Metal Cutting Tools Without Clearance, James Hartness.

Discussed by,

H. H. Suplee.

Interchangeable Involute Gear Tooth Systems, Ralph E. Flanders.

Discussed by,

Wilfred Lewis, L. D. Burlingame, D. F. Nisbet, Chas. W. Hunt, Oberlin Smith, W. Rautenstrauch, C. H. Logue, E. R. Fellows, Thos. Fawcus, F. de R. Furman, A. L. De Leeuw.

Spur Gearing on Heavy Railway Motor Equipments, Norman Litchfield.

Discussed by,

F. V. Henshaw, John Thomson, J. Kissick, Jr., Geo. Wm. Sargent, F. de R. Furman.

Luncheon was served to members and guests.

PROFESSIONAL SESSION

Thursday, 2 p.m., Auditorium

MISCELLANEOUS PAPERS

Articulated Compound Locomotives, C. J. Mellin.

Discussed by,

S. M. Vauclain, F. J. Cole, Harrington Emerson, L. R. Pomeroy, G. C. Fowler, G. R. Henderson, Alfred Lovell, G. H. Emerson.

Liquid Tachometers, Amasa Trowbridge.

Discussed by,

H. H. Waitt, H. G. Reist, W. F. Durand.

Training Workmen, H. L. Gantt.

Discussed by,

A. C. Humphreys, A. V. R. Scheel, Rudolf Roesler, T. F. Kelly, C. H. Buckley, H. K. Hathaway, Chas. Piez, C. N. Lauer, Lewis Sanders, J. C. Jurgensen, Willis E. Hall, Harrington Emerson, M. P. Higgins, Wm. Kent, J. M. Dodge.

Salt Manufacture, George B. Willcox.

Discussed by,

C. F. Hutchings.

Industrial Photography, S. Ashton Hand.
(Illustrated by Lantern Slides).

Discussed by,

C. J. H. Woodbury, H. B. Binsse, Chas. W. Hunt, H. H. Suplee, Ambrose Swasey.

Gas Power Section

Thursday, 2 p.m., Sixth Floor

SIMULTANEOUS SESSION

Business meeting and election of officers.

Reminiscences of a Gas Engine Designer, L. H. Nash.

Possibilities of the Gasolene Turbine, Prof. F. C. Wagner.

Discussed by,

S. A. Moss, C. E. Lucke.

RECEPTION

Thursday, 9 p.m.

The President and President-elect received the members and guests in the rooms of the Society. Dancing followed the reception. Supper was served from ten until twelve o'clock.

PROFESSIONAL SESSION

Friday, December 4, 9.30 a.m., Auditorium

EXPERIMENTAL DATA

Physical Properties of Carbonic Acid and the Conditions of its Economic Storage for Transportation, Prof. R. T. Stewart.

Discussed by,

J. C. Minor, Jr., H. E. Sturcke, Graham Clarke, L. H. Thullen, E. D. Meier, Sanford A. Moss, Wm. Kent.

THE SLIPPING POINT OF ROLLED BOILER TUBE JOINTS, Prof. O. P. Hood and Prof. G. L. Christensen.

Discussed by,

J. C. Parker, C. H. Benjamin, E. D. Meier, M. W. Sewall.

TESTS ON FRICTION CLUTCHES FOR POWER TRANSMISSION, Prof. Richard G. Dukes.

AN AVERAGING INSTRUMENT FOR POLAR DIAGRAMS, Prof. W. F. Durand.

COMMITTEES OF THE ANNUAL MEETING

COMMITTEE ON MEETINGS

CHARLES WHITING BAKER, *Chairman*

WILLIS E. HALL WILLIAM H. BRYAN L. R. POMEROY CHARLES E. LUCKE

LOCAL COMMITTEE

H. F. HOLLOWAY, *Chairman* CALVIN W. RICE, *Secretary*

L. R. ALBERGER	W. D. HOXIE	H. DE B. PARSONS
REG. R. BOLTON	F. R. HUTTON	N. B. PAYNE
G. H. BARBOUR	ALEX. C. HUMPHREYS	L. R. POMEROY
G. M. BASFORD	P. C. IDELL	GEORGE A. POST
C. W. E. CLARKE	H. S. ISHAM	FRED E. ROGERS
H. R. COBLEIGH	D. S. JACOBUS	H. W. ROWLEY
H. V. O. COES	R. F. JACOBUS	W. N. SAR VANT
JAMES V. V. COLWELL	W. A. JORDAN	F. A. SCHEFFLER
M. L. COOKE	E. B. KATTE	JESSE M. SMITH
W. C. DICKERMAN	W. J. KAUP	J. BEAUMONT SPENCER
F. E. EBERHARDT	WALTER C. KERR	ALBERT SPIES
ARTHUR FALKENAU	H. A. LARDNER	A. F. STILLMAN
A. M. FELDMAN	J. W. LIEB, JR.	F. H. STILLMAN
G. J. FORAN	FRED R. LOW	M. A. STONE, JR.
H. A. FOSTER	CHARLES E. LUCKE	H. G. STOTT
D. H. GILDERSLEEVE	E. W. MARSHALL	H. H. SUPLEE
WILLIS E. HALL	FRED J. MILLER	FRANK H. TAYLOR
F. A. HALSEY	E. P. MITCHELL	EDWARD VAN WINKLE
F. A. HANNAH	CHARLES A. MOORE	F. A. WALDRON
G. R. HENDERSON	WILLIAM NEWELL	W. H. WILEY
F. V. HENSHAW	W. W. NICHOLS	A. L. WILLISTON
CHARLES A. HOWARD	GEO. A. ORROK	IRA H. WOOLSON

HOTEL COMMITTEE

D. H. GILDERSLEEVE, *Chairman*

L. P. ALFORD	H. V. O. COES	W. P. PRESSINGER
GEO. H. BARBOUR	F.H. COLVIN	FRED A. SCHEFFLER
	R. E. FOX	

BUREAU OF INFORMATION

Albert Spies, *Chairman*

ASSIGNMENTS AT HEADQUARTERS FOR THE RECEPTION OF MEMBERS

Tuesday, December 1

Afternoon, 1-6	Evening, 7-10
D. H. Gildersleeve, *Chairman*	H. W. Rowley, *Chairman*
H. A. Foster	W. C. Kerr
H. S. Isham	W. P. Pressinger
Fred Rogers	H. H. Suplee
F. A. Scheffler	I. H. Woolson

Wednesday, December 2

Morning, 9 30-12	Afternoon, 1.30-6	Evening, 7.30-10
C. A. Moore, *Chairman*	G. M. Basford, *Chairman*	W. W. Nichols, *Chairman*
A. C. Humphreys	L. R. Alberger	F. A. Halsey
F. R. Hutton	A. Falkenau	D. S. Jacobus
H. de B. Parsons	G. J. Foran	
F. H. Taylor	W. D. Hoxie	
W. H. Wiley	Hosea Webster	

Thursday, December 3

Morning, 9.30-12	Afternoon, 1.30-6
J. W. Lieb Jr., *Chairman*	F. J. Miller, *Chairman*
Col. E. D. Meier	N. B. Payne
Jesse M. Smith	Albert Spies

Sub-Committee on Reception

Horatio A. Foster, *Chairman*

C. W. Aiken	A. M. Feldman	George A. Post
Edgar H. Berry	F. A. Hannah	Fred E. Rogers
Reginald P. Bolton	F. V. Henshaw	W. N. Sar Vant
C. W. E. Clarke	Charles A. Howard	J. Beaumont Spencer
H. R. Cobleigh	P. C. Idell	A. F. Stillman
H. V. Coes	R. F. Jacobus	F. H. Stillman
M. L. Cooke	W. A. Jordan	M. A. Stone, Jr.
W. C. Dickerman	W. J. Kaup	Edward Van Winkle
W. N. Dickinson	E. W. Marshall	F. A. Waldron
F. E. Eberhardt	William Newell	A. L. Williston

Friday, December 4

Morning, 9.30-12

Fred R. Low, *Chairman*	G. R. Henderson	E. B. Katte

PRINTING COMMITTEE

H. V. Coes, *Chairman*

C. A. Howard
Albert Spies
Mason A. Stone, Jr.
J. V. V. Colwell
D. H. Gildersleeve

EXCURSION COMMITTEE

James V. V. Colwell, *Chairman*

E. B. Katte
W. H. Hulbert
H. W. Rowley
F. L. DuBosque
B. M. Mitchell

LADIES' RECEPTION COMMITTEE

Mrs. Jesse M. Smith, *Chairman*

Miss S. Edna Johnston, Associate Editor Am. Soc. M. E., *Secretary*

Mrs. H. C. Abell	Mrs. H. F. Holloway	Mrs. Eugene Price
Mrs. Charles W. Baker	Mrs. G. S. Humphrey	Mrs. Calvin W. Rice
Mrs. G. H. Barbour	Mrs. C. W. Hunt	Mrs. E. M. Sanderson
Mrs. A. R. Baylis	Mrs. F. R. Hutton	Mrs. J. P. Sneddon
Mrs. E. H. Berry	Mrs. J. E. Jones	Mrs. Stevenson Taylor
Mrs. Edward Ciordi	Mrs. J. A. Kinkead	Mrs. H. G. Torrey
Mrs. J. Van V. Colwell	Mrs. G. L. Knight	Mrs. S. E. Whitaker
Mrs. H. E. Cook	Mrs. J. W. Lieb, Jr.	Dr. Lucy O. Wight
Mrs. G. K. Fowler	Mrs. Fred R. Low	Mrs. A. L. Williston
Mrs. D. H. Gildersleeve	The Misses Meier	Mrs. T. E. Wilson
Mrs. F. A. Hall	Mrs. C. W. Obert	Mrs. Ira H. Woolson
	Mrs. G. A. Orrok	

ACCOUNT OF THE ANNUAL MEETING

The twenty-ninth annual meeting of The American Society of Mechanical Engineers was held in the Engineering Societies Building December 1–4, with an attendance of 738 members, the largest in the history of the Society, and a total registration of 1048. This total is not as high as that of a year ago, but it actually represents a larger attendance, since this year only those were registered as guests who were to participate in the social functions.

The meeting was noteworthy in several particulars. There were six professional sessions, one more than at any previous convention. One of these was a machine shop session which led to the first steps being taken toward the formation of a machine shop section. The subject of aëronautics was considered for the first

time in America by a national engineering society. On Tuesday evening, besides the President's address, was the conferring of honorary membership upon Dr. John A. Brashear, followed by his delightful lecture upon A Journey among the Stars. On Wednesday evening was the intensely interesting lecture by Lieut. Frank P. Lahm on The Conquest of the Air.

OPENING SESSION, TUESDAY EVENING

The opening session of the convention is always an anticipated occasion. It is the social event where members first greet one another and is, moreover, the time for the delivery of the annual address of the President.

The session was called together by President M. L. Holman, who proceeded at once with his address. In view of the activity of the Society in the conservation movement and the participation by President Holman in the Governors' meeting at Washington, his subject was very properly The Conservation Idea as Applied to The American Society of Mechanical Engineers. The address is printed in full in this volume.

Following the address, Mr. W. R. Warner, Chairman of the Committee appointed for the purpose, presented Dr. John A. Brashear for honorary membership, saying:

It is my privilege to present John Alfred Brashear, Sc.D., LL.D., F.R.A.S., Member of the Astronomical Societies of Great Britain, France and Belgium and of the American Philosophical Society; Past Chancellor of the University of Western Pennsylvania; Organizer of Carnegie Institute; Collaborator with Langley in devising and making the bolometer for measuring heat to 1/100 000 of a degree; Co-worker with Morley and Michaelson in constructing the interferometer which established wave lengths of light as the unit of linear measure; Maker of astronomical instruments of unequaled delicacy and precision, the spectroscope whereby Keeler discovered the constitution of Saturn's rings and Campbell the motion of the stars in the line of sight, and the spectroheliograph with which Hale has determined the constitution of the sun and analyzed its elements; Physicist, Astronomer, Educator, whose contributions to science and technical learning are surpassed in value only by his personal worth and greatness of soul, which have endeared him to all who know him.

Dr. Brashear is thus formally presented in order that he may receive from you the certificate of honorary membership in The American Society of Mechanical Engineers.

The President then replied:

John Alfred Brashear, Eminent Engineer, Scientist, Astronomer, Educator, and Craftsman, the Society is proud to honor you with the distinction I am

now to confer. The mechanical engineering profession recognizes your achievements in the production of parts of optical apparatus whose mechanical perfection has never been approached, and which it will be difficult to surpass. For these reasons, by the authority conferred upon me, I now advise you that you have been elected to Honorary Membership in The American Society of Mechanical Engineers, and to all the rights and privileges attaching to this distinction.

In Witness Whereof, will you accept at our hands the diploma of such membership.

Dr. Brashear had been invited to address the Society on A Journey Among the Stars, and his lecture followed. By way of introduction he said that no honor which had been conferred upon him came so close as that just given—an honor by men who know the worth of real work. There is something that draws men to one another which is elevating and ennobling to the man who loves his work for its own sake. He said, "There is a beautiful adage that I learned long ago. It goes this way: 'What man is there, who, coming in contact with great souls, is not made happier and better thereby? A drop of water on the leaf of a lotus glitters with the luster of a pearl.' And so our deeds may be small, if they are only done in the right spirit."

The speaker then took his audience on a journey made possible only by the wonderful developments of stellar photography, since the camera alone can penetrate the universe and the infinite spaces beyond. The photographs which Dr. Brashear displayed were as wonderful as those shown by him at Detroit at the time of the Spring Meeting. Striking features of the lecture were the illustrations used to indicate the magnitude of the heavens and to show how the movement of stars is determined by means of the spectrum.

BUSINESS MEETING

On Wednesday morning was the annual business meeting and the first professional session. President Holman called the meeting to order and reminded those present that in view of the large number of papers to be presented at the various sessions, it might be necessary for the Chair to limit discussions upon papers to the time specified by the rules. He suggested, however, that as there were eight auditoriums in the building, it would be possible to hold simultaneous meetings if any considerable number of members desired to continue the discussion upon a given topic beyond the time allotted. This plan was actually carried out in order to extend the discussion upon the papers given on the following morning upon machine shop sub-

jects. On the afternoon of that day there were three sessions in progress at one time.

The first order of business was the reading of the report of the Tellers of Election. There were 168 applicants for membership and 22 for advance in grade. These having been duly balloted were declared elected and the names are published in the Appendix to the Annual Report of the Council and Committees in this volume.

The following officers were declared elected for the succeeding year: President, Jesse M. Smith; Vice-Presidents, Geo. M. Bond, Prof. R. C. Carpenter, F. M. Whyte; Managers, H. L. Gantt, Will J. Sando, I. E. Moultrop; Treasurer, Maj. Wm. H. Wiley.

The newly elected President, Jesse M. Smith, was then escorted to the Chair and spoke briefly as follows:

The honor which you have conferred upon me I sincerely appreciate. I recognize also the great responsibility that comes with the honor. The influence of the Society in the advancement of engineering, must not only be maintained, but greatly extended. It will be my endeavor to conserve the best traditions of the Society and to aid in its taking a position still further forward.

Were I to follow precedent I would say no more at this time; but there is a subject which seems to me of great importance to the future of the Society, which I would like briefly to call to your attention.

The prominent new feature of the constitution, under which the Society has been operating for five years, is the formation of standing committees to foster, organize and direct its various activities. Each of these seven committees is composed of five members. One member retires at the end of each year and a new member is appointed by the President. Thus the committees, while permanent as organizations, are being constantly and automatically renewed as to personnel.

The Secretary of the Society is the secretary to the Council and also the secretary to each of the standing committees. Thus the membership, the Council, and the various committees are properly coördinated to work together in harmony for the advancement of the Society; each committee taking charge of, and being responsible for, the particular work assigned to it by the Constitution. The function of each standing committee is to initiate, promote and organize the work given to it by the Constitution and By-Laws, subject to the approval of the Council.

It is evident that when all the committees are in full operation, the Secretary as the executive officer will be fully occupied in carrying out the various activities initiated and organized by these various standing committees. The Secretary should not be called upon to do work which properly belongs to the committees, however willing he may be to do so.

The orderly and systematic work of the Secretary is of the greatest importance. Through the good work of his office, the membership, scattered over the world, receives prompt, accurate and full information of what the Society is doing for the advancement of engineering.

Great and important and good as may be the work of the Secretary, there is a greater and different work to be done by the standing committees; and it is their

active work which will inspire greater activity among the membership of the Society.

These committees, when fully organized, will contain 35 men who should be selected from the membership because of their peculiar qualifications and fitness to do the special work required. They must be men who are able and willing to devote the necessary time to the work. These 35 men are officially called to places of honor. They should, and undoubtedly will, respond to the call with ardor, and with the determination to "set the pace" for the membership at large in the march forward. The success and influence of the Society should be in direct proportion to the number of men who are active in its welfare.

The good work of the standing committees is already well commenced, but it should be extended and expanded, and improved with the experience of years and by the infusion of new and young blood until each committee accomplishes its full function.

The best work done in the Society has been done by men inspired with love for the profession of engineering. Love for the profession does not die. There are now, and always will be in this Society, men who are thus inspired. Let them come forward and take up the work before them, or if they be diffident, let us seek them out and bring them forward.

Let the members of the committees bring their work up to such a high degree of excellence and efficiency, that a position on a standing committee of The American Society of Mechanical Engineers will be an honor which a rising engineer, who loves his profession, will wish to attain.

FIRST PROFESSIONAL PAPERS

After the remarks by the President-elect, came the two professional papers of the morning, the first one being The Engineer and the People—a Plan for a Larger Measure of Coöperation between the Society and the General Public, by Morris Llewellyn Cooke, Philadelphia, Pa., published in this volume. This paper advocates a more direct interest by engineers in affairs affecting the public and a greater effort to enlighten the public as to the advantages and achievements of engineering. A corresponding change is already in progress in other professions which the engineering profession may well emulate. The author asked for the appointment of a standing committee to be known as the Committee on Relations with the Public.

This paper was largely discussed both by members of the Society and by prominent men identified with other lines of activity.

Mr. Ambrose Swasey offered the following resolution, which was seconded by Mr. Fred. W. Taylor:

Resolved: That we recommend to the Council the appointment of a professional committee to advocate, consider and report on the methods whereby the Society may more directly coöperate with the public on engineering matters and on the general policy which should control such coöperation.

The resolution was unanimously carried.

Prof. F. R. Hutton stated, in connection with the recommendation of Mr. Cooke, that a Committee on Relations with the Public should be created, that this would require an amendment to the Constitution. He therefore gave notice, in accordance with the provision of the Constitution, of the purpose to make such an amendment to Article C45 at the Spring meeting, at which such amendment can come up for discussion.

The second paper of the morning was by Major Geo. O. Squier of the Signal Corps, U. S. A., on The Present Status of Military Aëronautics. This paper with its illustrations was published in full in the December number of The Journal and is a remarkably complete statement of the development of the leading types of air craft. It deals to a certain extent with the problems of design and gives dimensions of different dirigible balloons and aëroplanes, besides data upon the construction of these types of machines.

WEDNESDAY AFTERNOON SESSION

The meeting was called to order by President Holman who requested Mr. Geo. R. Stetson to preside.

The first paper was upon A Method of Obtaining Ratios of Specific Heat of Vapors, by Mr. A. R. Dodge of Schenectady, N. Y. This outlines a method of obtaining the ratio of specific heat which does not involve the use of available steam tables nor a condition in which the steam is presumed to be without moisture or superheat. Tables of data are included. The paper was discussed by Dr. Harvey N. Davis, who had had an opportunity to go over the data previous to the presentation of the paper in connection with some of his own work and believed the method of using the throttling calorimeter which Mr. Dodge had devised was a great advantage in technique in the measuring of superheat.

The next paper was by Dr. Harvey N. Davis, Cambridge, Mass., upon The Total Heat of Saturated Steam. It has, for some time, been thought that Regnault's formula for the total heat of saturated steam is considerably in error. This conclusion is confirmed by computing the value from the results of various experimenters upon the specific heat of superheated steam. From their work Dr. Davis has deduced a formula which is believed to give much more accurate results.

Following these papers upon what may properly be classed as

engineering physics dealing with the refinements of engineering, were two papers by Mr. C. R. Weymouth of San Francisco, Cal., presenting important results in the line of engineering practice. These were presented for the author by Prof. D. S. Jacobus. The subjects were Fuel Economy Tests at a Large Oil Burning Electric Plant and Unnecessary Losses in Firing Fuel Oil. The first of these contains results of tests upon a 15 000 kw. power plant of the Pacific Light and Power Co., Redondo, Cal., having steam engine prime movers and using crude oil as fuel. The results were given for various uniform loads, ranging from 2000 to 5000 kw.; also for the entire station on a variable railway load.

The second paper describes apparatus for securing proper adjustments in automatic firing for steam boilers in plants burning liquid fuel. The paper also has valuable data upon the heat value of California oils.

WEDNESDAY EVENING LECTURE

On Wednesday evening was the lecture on aëronautics by Lieut. Frank P. Lahm of the Signal Corps, U. S. Army, who took as his subject The Conquest of the Air. The interest aroused by this feature of the annual meeting is indicated by the size of the audience which greeted the speaker. It was the largest in the history of the Engineering Societies' Building. Every seat and the available standing room of the auditorium were taken and the evident anticipation of the audience changed to enthusiasm as the speaker skilfully developed his subject.

The lecture was illustrated by a profusion of lantern slides and by motion pictures of dirigible balloons and the Wright Brothers' aëroplane. The speaker traced the important events related to the development of air ships during the past 125 years, beginning with the discovery in France of the principle of the hot-air balloon by the Montgolfier brothers. Benjamin Franklin witnessed one of the earliest ascensions in France by hot-air balloon and is reported to have had faith in this type of craft. In reply to the question "Of what use are balloons?" he said "Of what use is a new-born babe?"

Lieutenant Lahm is an experienced balloonist and his graphic description of a balloon ascension was so evidently a recital of his own experiences as to add force to his remarks. He described the process of preparing a balloon for ascension and inflating it and the instruments used by the aëronaut. To maintain equilibrium when

a balloon is in the air requires close attention. A cloud passing across the sun cools the gas and starts the balloon down, or a burst of sunshine on a cloudy day produces the opposite effect. The cool air encountered in passing over a forest has the same effect as the cloud. The pilot must know at once when his balloon starts up or down. A little sand thrown out at the beginning of a descent will do more to stop it than a large quantity later. The registering barometer does not record quickly enough, so an aneroid barometer with a circular dial and a needle is used; or more often a statoscope, which indicates instantly whether a balloon is going up or down. A sextant with artificial horizon is used for finding the latitude of a balloon when above the clouds.

Answering the questions that are commonly asked about ballooning, the speaker said there is nothing by which one can measure his height, and there is no unpleasant motion as in an elevator. Suspended in the air and moving with it, one does not realize that he is moving at all. There is therefore no unpleasant sensation; the delights of ballooning can be realized in no other way; and at the conclusion of a trip the landing is made, the balloon shipped back to the starting point by freight or express, and "its passengers settle down comfortably to their dinner in the dining car and go over again the enjoyable incidents of the ascension."

Interest in aëronautics was greatly stimulated by the international competition for the Gordon Bennett Cup in 1906. This race was from Paris and was won, the speaker modestly said, by "an American." This race was with free balloons, but it is expected that dirigibles will enter future races.

Descriptions were given of representative dirigible balloons, beginning with that of Santos Dumont in 1898, who succeeded three years later in circling the Eiffel Tower in his balloon. Another early and successful airship was that built by the Frenchman Julliot, an engineer in the employ of Lebaudy brothers, wealthy sugar refiners in Paris, who backed him. His efforts finally led to the construction of the well-known *Patrie.* Many views were shown of the familiar war balloons of the various countries, including the *République, Ville de Paris, Gross, Parseval, Zeppelin* and the dirigibles of the British and American armies which bear numbers only in place of names.

Of greater interest than any of these, perhaps, was the account of the Wellman airship designed for reaching the North Pole. It was built in Paris in 1906 and transported to Spitzburgen. It was designed

as a weight carrier, with a speed of only 15 miles an hour, and would carry a crew of three men, dogs, sleds, boat, and abundant provisions. By taking advantage of favorable winds it was hoped to cover the 700 miles to the pole in two days. The return journey would be less difficult as any direction would lead back to civilization. In attempting its flight, however, bearings were lost and the ship had to be returned to its quarters.

In dealing with heavier-than-air machines prominence was given to the work of the Wright brothers, whose aëroplane at Fort Myer, Va., established so notable a record. Proper recognition was also given to the early work of Professor Langley who, in 1896, constructed a model that flew a mile under steam power.

The culminating feature of the lecture was the moving pictures. Huge dirigible balloons were seen slowly moving out of their balloon houses and returning. The Baldwin dirigible was shown in actual flight and there were views of the rapidly moving Wright brothers' machine, which flitted across the screen like a great bird.

In conclusion the speaker said, "With dirigible balloons capable of remaining in the air 13 hours, covering a distance of 176 miles; with the Wright aëroplane which has already remained in the air an hour and a half and has carried two persons at the rate of 40 miles an hour, under the perfect control of its operator, I think it will be agreed that the experimental stage is past and the conquest of the air is a fact."

THURSDAY MORNING—MACHINE SHOP SESSION

The session on Thursday morning was devoted to papers upon machine shop practice. Vice-President Fred J. Miller presided.

The first paper was upon Efficiency Tests of Milling Machines and Milling Cutters, by A. L. DeLeeuw, Cincinnati, O. It pointed out the desirability of indicating the power of a machine tool by the amount of metal which it is capable of removing, rather than by the size of the driving pulley and belt. It described tests upon several milling machines for the purpose of ascertaining the amount of metal removed and the capacity; also the horsepower required under various conditions of feed and speed. It considered the mechanical efficiency of the machines and gave results of tests showing the importance of improvement in milling cutters.

The next paper was upon the Development of the High Speed Milling Cutter with Inserted Blades for High Speed Steel, by Wilfred

Lewis and Wm. H. Taylor, both of Philadelphia, Pa. The milling cutter which formed the basis of this paper has inserted helical blades of high speed steel, mounted in a steel holder to give a solid backing for the blades on the driving side against which they are held by a soft metal filler on the opposite side. The cutting power of a cutter built up in this way is so great that it is stated to be beyond the capacity of any milling machine now on the market. Tables of results of tests were included in the paper.

Following these papers upon miiling practice was one upon lathe tools by James Hartness, Springfield, Vt., entitled Metal Cutting Tools without Clearance. The tool operates on a new principle developed by the author which, contrary to the universal plan of cutting tools, was designed to be used without clearance. The tool is supported in a holder so constructed as to allow a slight oscillatory motion which permits the face of the tool to bear against the face of the metal from which the chip is being cut. This steadies the tool, prevents lateral vibration, which is detrimental to the cutting edge of any tool, and so permits a more acute cutting edge.

In presenting the paper, Mr. Hartness prefaced the reading by remarks upon the use of lubricants, stating that it was necessary in certain cutting operations to use lard oil. The paper was discussed by Henry Harrison Suplee who mentioned that he had used wood cutting tools without clearance in planing machines for much the same purpose as Mr. Hartness used his metal cutting tool without clearance. The objection for that class of work, however, had been the heating of the cutting edges.

The last two papers were upon the subject of gearing, which in this instance, as in times past, proved to be most prolific of discussion. The first paper, entitled Interchangeable Involute Gear Tooth Systems, by Ralph E. Flanders of New York, showed the effect of varying the pressure angle and height of addendum on the various practical qualities of gearing, such as continuous action, side pressure, strength, efficiency, etc. The author asked for a discussion on the question of the appointment of a Committee to investigate and report as to the wisdom of an alternative form of gearing for heavy use. The first discussion was by Wilfred Lewis, who advocated the investigation of the subject and made the following motion:

"I would therefore propose that this subject be referred, as Mr. Flanders suggested, to a committee of the Society to investigate and report upon the adoption of a standard system of involute gearing. The paper covers the case from a 12-tooth pinion to a rack.

That is as far as I would go with such a system. If internal gears are employed, that would be understood to be more or less special."

The second paper upon gearing was upon Spur Gearing on Heavy Railway Motor Equipments, by Norman Litchfield, New York. This dealt with the breakage of gearing in heavy electric railway service and referred to the work of the Interborough Rapid Transit Co. in overcoming this difficulty. It considered the materials and design of gearing for heavy duty including the shape of tooth outlines employed.

The discussion upon the gearing papers was so extended that the session adjourned until afternoon, at which time Prof. W. Rautenstrauch presided.

At this session the resolution offered by Mr. Lewis in the morning calling for the appointment of a committee to consider the matter of interchangeable involute gearing was brought up and unanimously carried. In order to conform to the usual procedure in such matters it was suggested by Mr. Elmer H. Neff that the resolution should be in the form of a request to the Council to take up the matter, which was concurred in by Mr. Lewis.

THURSDAY AFTERNOON SESSION

Prof. F. R. Hutton, who acted as Chairman, announced that the paper by Mr. Mellin upon Articulated Compound Locomotives would be deferred until later in the afternoon when the lantern slides would be more effective.

The first paper to be presented was upon Liquid Tachometers, by Amasa Trowbridge, Hartford, Conn. This described the Veeder liquid tachometer and methods used in testing and calibrating it.

The next paper, by H. L. Gantt, Pawtucket, R. I., upon Training Workmen in Habits of Industry and Coöperation, drew out the most active discussion of any paper of the convention. It emphasized the fact that with the advent of the scientifically educated engineer capable of substituting a scientific solution of problems for the empirical solution of the mechanic, the responsibility of training workers actually shifts to his shoulders. If he properly conducts this training along the lines of scientific investigation, efficiency of the workmen can be so greatly increased that the employer can afford to pay far in excess of the compensation usually allowed.

The discussions were very favorable to the methods of Mr. Gantt, since they represented humanitarian ideas rather than the purely

commercial aspect so often predominating in the training of workmen.

Geo. B. Willcox of Saginaw, Mich., next presented a paper on Salt Manufacture, in which he described apparatus used in the manufacture of salt by the grainer process including the design of evaporated grainers built of reinforced concrete, and devices for handling and conveying salt and loading salt barrels into cars.

The next paper, upon Industrial Photography, by S. Ashton Hand, Cleveland, O., was illustrated by lantern slides showing some remarkably perfect results obtained in photographing machinery and interior of shops. These illustrated methods used by the author in bringing out details and avoiding shadows or too prominent high lights. He showed by a series of plates results that could be obtained in developing plates that were under or over-exposed; the effect of different lengths of exposure upon plates and how certain defects in plates could be remedied.

A paper upon Articulated Compound Locomotives, by C. J. Mellin, Schenectady, New York, which had been deferred until the end of the meeting, described locomotives articulated by the Mallet method by means of which the tractive power can be doubled over that of an ordinary engine for a given weight of rail with a substantial saving in fuel.

GAS POWER MEETING

The Gas Power Section held a meeting on Thursday afternoon, with Dr. Charles E. Lucke in the chair, and about 150 members of the Section present.

The executive committee of the Section reported that it had proceeded to the election of officers for the ensuing year, as follows:

Chairman, F. R. Low; Secretary, George A. Orrok; Chairman of the Membership Committee, Robert T. Lozier; Chairman of the Meetings Committee, Cecil P. Poole; Executive Committee, F. H. Stillman, George I. Rockwood, R. H. Fernald, F. R. Hutton, H. H. Suplee.

The retiring executive committee recommended the appointment of a nominating committee, to place at least two candidates in nomination for each office; a committee on installations, to keep a list of all power plants, giving complete data as to equipment; a committee on plant operation, to collect information as to load characteristics, costs of operation, behavior of apparatus, etc.; and a com-

mittee on breakdowns, failures, etc., to collect and file information as to accidents, unsatisfactory operation, etc.

In connection with the work of the standardization committee, a communication from Prof. Lionel S. Marks was presented by Prof. Ira S. Hollis, discussing the high and low heat values, and calling attention to the fact that the "effective" heat value in German practice is the heat value of the gas under the conditions of temperature and pressure at which it is used.

An interesting paper was presented by Mr. L. H. Nash, reviewing his own experiences in gas-engine work, and showing the extent to which old ideas have cropped up from time to time. This paper, which was profusely illustrated by lantern slides, was most interesting and instructive. A paper upon Some Possibilities of the Gasolene Turbine, by Prof. Frank C. Wagner, was read in abstract, in the absence of the author. This paper discussed analytically the relative effects of an excess of air and an injection of water for keeping down the temperature of the gases.

Dr. Lucke called attention to the observed facts as to the behavior of the free expansion of gases in nozzles, showing that assumed effects are not realized in practice, and recommended further experimentation in this direction.

THURSDAY EVENING RECEPTION

On Thursday evening was the annual reception, which was held in the rooms of the Society and followed by a collation and dancing.

FRIDAY MORNING SESSION

The papers for Friday morning presented mainly the results of tests upon various types of apparatus. The first, by Prof. R. T. Stewart of Pittsburg, Pa., was one of a series of papers which he has read before this Society upon extended lines of investigation which he has conducted. This paper dealt with the Physical Properties of Carbonic Acid and the Conditions of its Economic Storage for Transportation. It treated exhaustively of the physical properties of carbonic acid, and mentioned tables of data heretofore unavailable. There were also suggestions for the design of cylinders to withstand high pressures. The discussion on the paper was, from the nature of the subject, somewhat out of the sphere of mechanical engineering and bore largely upon the safety of carbonic acid cylinders used for the storage and transportation of a chemical.

The Slipping Point of Rolled Boiler Tube Joints, by Professors O. P. Hood and G. L. Christensen, Houghton, Mich., was the title of the next paper which had for its object the recording of data regarding the behavior of joints made by rolling boiler tubes into the containing holes of the tube sheets.

The third paper, by Prof. R. G. Dukes on Tests on Friction Clutches for Power Transmission, gave results of tests upon friction clutches of different makes.

The final paper of the morning was a brief description, with the theory of design, of An Averaging Instrument for Polar Diagrams, by Prof. W. F. Durand, Stanford University, Cal. It was intended to supply information for the design of the planimeter for use on polar diagrams so commonly used on different types of recording instruments.

The meeting closed with the following resolutions:

WHEREAS: The American Society of Mechanical Engineers at its Annual Meeting, December, 1908, desire to express its appreciation to those who have provided opportunities for entertainment; and on behalf of its visiting members thanks for the welcome so cordially given by the local members and their friends of New York and vicinity.

Be it Resolved, that the Secretary be instructed to extend the thanks of the Society and to express the appreciation of its members and guests to the local committees for their untiring efforts; to those who have sent invitations to visit engineering works and places of interest; to Professor Brashear for his delightful lecture; to Brig. Gen. James Allen and his associates of the Signal Corps for the remarkable presentation of the subject of aëronautics before the Society; and especially to the ladies who extended so generous a hospitality to their guests.

No. 1207

THE ANNUAL REPORTS OF THE COUNCIL AND COMMITTEES, 1908

REPORT OF THE COUNCIL

The Council submits herewith a report of the activities of the year, together with the reports of the standing and special committees.

One of the most extensive and far-reaching movements of the year has been the Society's participation in the National movement for the conservation of natural resources, in response to the invitation of the President of the United States. Under the heading Conservation published in this report, is given a complete account of the Society's participation in the Congress of the Governors of the States, and in the account of the April meeting of the Society in this volume will be found abstracts of the addresses made at a special meeting of the Society on Conservation.

Understanding the need for extending the influence and service of the Society, the Council has provided for the formation of professional sections and student branches. Other situations are being met as fast as practicable. Professional sections foster the development of special branches of engineering in which members and others not eligible to membership are engaged, and enable those dealing with a special branch of work to devote more time to the discussion of their problems than the Meetings Committee, in the interest of the Society, would feel justified in allowing at the regular meetings. Student Branches establish relations with students in accredited engineering schools; and encourage students to hold meetings of their own and to attend the meetings of the Society. By reason of their affiliation to the Society, they receive The Journal. The plan for Affiliated Societies was referred to a special committee composed of F. R. Hutton, *Chairman*, R. H. Fernald, F. W. Taylor and H. H. Suplee, who presented rules for the formation and government of such sections which were approved by the Council. These rules are published under "Amendments" in this report.

The Council decided that all material appearing in The Journal may be immediately republished without restriction. Heretofore

papers have not been released until the date of the meeting at which they were to be presented. The usual footnote to this effect, which has appeared with each paper, will now be omitted, and the press throughout the world are invited to republish at once the papers appearing in The Journal, in part or in whole. By this means, the widest possible publicity will be given to the contributions of the various authors, and greater interest in the meetings will be awakened among engineers, particularly if interested in any special subject to come up for discussion. It is hoped that the free and unrestricted distribution of the material contributed to the Society will redound to the benefit both of the Society and of the profession. It is believed that this step will meet with general approval.

A conference for the international standardization of pipe threads was held in Paris, France, June 23, under the auspices of the Société Technique de l'Industrie du Gaz. In response to an invitation, this Society appointed a special committee which consisted of E. M. Herr, *Chairman*, George M. Bond, Wm. J. Baldwin and Stanley G. Flagg, Jr., and appointed Laurence V. Benet of Paris, as a special representative at the conference. The report of the committee was forwarded to M. Benet for presentation at the conference.

The Society was represented by the Honorary Vice-Presidents at the following events.

The memorial services for Lord Kelvin held under the auspices of the American Institute of Electrical Engineers, January 12: Andrew Carnegie, Thomas Edison, Rear-Admiral Melville, Geo. Westinghouse and Benjamin F. Isherwood. Funeral services of the late Coleman Sellers, Past-President of the Society, December 31: J. M. Dodge and F. W. Taylor. The installation of Prof. W. F. M. Goss as Dean of the College of Engineering of the University of Illinois, February 5: Wm. Forsyth. The dinner of the Society of Civil Engineers of Boston, March 10: Fred J. Miller. The meeting of the Society of Automobile Engineers, March 10: Prof. F. R. Hutton. The luncheon given by President Macdonald of the American Society of Civil Engineers to the Hon. Gifford Pinchot, March 5: Charles Whiting Baker. The meeting of the International Congress of Navigation, St. Petersburg, May 31–June 7: W. E. Smith. To represent the Society and its President at the conference of the four national engineering societies on the conservation of our national resources, May 9: Prof. F. R. Hutton, Honorary Secretary. The dedication of the new buildings of the College of the City of

New York, May 14: Prof. Ira H. Woolson. The convention of the National Electric Light Association in Chicago, May 18: George M. Brill. The International Congress of Refrigerating Industries in Paris, September 17–23: Gardner Tufts Voorhees. The Anniversary Banquet of the Société des Ingenieurs Civils de France, May 16: Lawrence V. Benet and Gustave Canet. Forest Festival, Biltmore Estate, November 26, 27, 28: Charles E. Waddell. American Mining Congress, December 2–5: W. M. McFarland, W. A. Bole. National Irrigation Congress, September 29–October 3: W. B. Gregory.

The following deaths are reported:

Coleman Sellers, Past-President, Thos. Fitch Rowland, Geo. W. Hammond, A. G. Goldthwait, Wm. S. Love, W. H. Hume, A. F. Knight, Edw. B. Brisley, O. F. Nichols, W. Roberts, James Powell, W. H. Wiggin, C. H. L. Smith, C. D. Pierce, Gustave Herrmann, Hon. Mem., F. C. Warman, F. N. Fowler, Fredk. A. Johnson, B. F. Schaefer, Ferdinand Phillips, S. B. Cox, Jos. Stone, F. B. Kleinhans, F. McGowan, Wm. Anson Pearson, W. H. Bailey, H. F. Glenn, Gustave Canet, Hon. Mem. F. A. C. Perrine, and E. G. Eberhardt.

The following resignations are reported:

Alex. Delaney, C. W. Rowe, W. L. Clements, Geo. A. Ensign, Geo. B. Bartlett, W. L. Hedenberg, Jno. W. Loveland, M. T. Conklin, G. L. Backstrom, Willard L. Case, Jno. B. Fleming, Geo. H. Lilley, W. C. Swift, P. P. Rooney, J. M. Barnay, Lemuel Clark, H. deF. Hubbard, J. L. Pitkin, H. Van Atta, J. C. Knight, C. E. Bement, J. R. Caldwell, Walter Kirton, Wm. M. Power, Jos. D Wallace, Jos. Kuhn, Frank G. Brown, Jr., Donald Enock, Camille A. Lamy, C. Reeve, John Dick, W. S. Auchincloss, C. E. Brown, Alex. Gordon, Morris M. Green, E. B. Guthrie, E. E. Hanna, C. C. King, F. G. Kretschmer, A. C. Linzee, J. H. Massie, Fredk. McIntosh, A. T. Porter, Justin A. Ware, Max H. Wickhorst, Alfred Marshall, J. F. Wilcox, James Inglis, B. J. Dashiell, Louis C. Schaeffer, O. V. DeGaigne, and H. S. Richardson.

During the year the Society has received, through the kindness of Mrs. Westinghouse, an excellent portrait of George Westinghouse, Hon. Mem. Am. Soc. M. E., which has been placed in the Council room.

ELECTION OF JOHN A. BRASHEAR TO HONORARY MEMBERSHIP IN THE SOCIETY

At the meeting of the Council Tuesday, November 10, Dr. John A. Brashear, of Allegheny, Pa., was unanimously elected Honorary Member of the Society, in response to a petition signed by the following members:

W. R. Warner
C. F. Brush
John Fritz
Chas. H. Morgan
Victor E. Edwards
C. M. Schwab
W. M. McFarland
Alexander Taylor
E. S. McClellan
George I. Alden
Joseph F. Klein
Chas. Wallace Hunt
Jesse M. Smith
Geo. W. Melville
Henry L. Barton
Wm. A. Bole
Walter C. Kerr
H. H. Westinghouse

Dr. Brashear is an expert in the manufacture and development of astronomical and physical instruments of precision and an acknowledged leader in astronomical research.

It is the spirit of the Council to favor the exchange of courtesies with other Societies, and it accordingly directed that the Secretary send invitations to the secretaries of the Institute of Civil Engineers (London), the Institute of Mechanical Engineers (London), the Institute of Electrical Engineers (London), and the Iron and Steel Institute (London), offering the freedom of the Society's headquarters and library to their members visiting in America.

The Pratt & Whitney Co. presented to the Society a set of standard gages made by them in conformance with the standards for machine screws which this Society's committee recommended. The Council accepted the gages, to be used as a standard and authentic set, and gave a vote of thanks to the Pratt & Whitney Co.

Several years ago, a committee consisting of Messrs. John Fritz, Stephen W. Baldwin, R. C. Carpenter, Walter C. Kerr, E. A. Uehling, William Hewitt and Gus. C. Henning, collected $788.32, toward a memorial to Dr. Robert Henry Thurston, first President of the Society. This money was never expended, for the reason that a suitable memorial was not conceived.

The Committee transferred the amount to Prof. F. R. Hutton as Treasurer, and Professor Hutton, June 19, 1908, transferred it to the Society.

Upon the unveiling of the tablet at Cornell University erected to Dr. Thurston's memory by the alumni and engineering students of Sibley College at the time of Dr. Thurston's death, the idea occurred to the members of the Society that a replica of this tablet would be a suitable memorial for the Society. Permission was obtained from the Sibley Alumni Committee and H. A. McNeil, the sculptor of the Cornell tablet, who was a personal friend of Dr. Thurston, was secured

for the work. The Council voted, Oct. 13, 1909, that the sum previously collected, \$797.17, be expended and voted the additional sum of \$52.83 necessary to cover the cost of the tablet, \$850.

The Council subscribed to the Institution of Civil Engineers of Great Britain \$155 toward a window to be placed in Westminster Abbey to the memory of Sir Benjamin Baker, Hon. Mem. Am. Soc. M. E. This recognition has been given to engineers in but few instances, being restricted to a statue of Telford, a bust of Watt, and memorial windows to Robert Stephenson, Joseph Locke, and Sir William Siemens. The Institution will place a similar window in their new home.

The Council presented to Prof. F. R. Hutton, Honorary Secretary, Am. Soc. M. E., engrossed resolutions in acknowledgment of long service to the Society.

The Council voted that the membership in other bodies of the Society, as such, be terminated.

The Council appointed a committee composed of F. R. Hutton, G. M. Basford and the Secretary, to devise a plan whereby the publications of the Society might be developed, and approved the recommendations of this committee to advance \$10 000 for this purpose and to appoint an editor and an advertising manager. Lester G. French was appointed Editor, and E. J. Gibling Advertising Manager.

Dr. F. R. Hutton was reappointed trustee of the United Engineering Societies for a term of three years.

The Council appointed John W. Lieb, Jr., and F. W. Taylor a special committee to consider the revision of the present code for conducting steam boiler trials.

In response to an invitation from the American Association for the Advancement of Science to each of the National Engineering Societies, the Council appointed Alex. C. Humphreys and Fred J. Miller, as representatives of this Society, upon the Council of that Association.

Upon the recommendation of the Gas Power Section, the Council appointed Prof. C. E. Lucke, *Chairman*, Prof. D. S. Jacobus, Messrs. Geo. H. Barrus, C. N. Scott and E. T. Adams, a Committee on the Revision and Extension of the Code for Testing Gas Power Machinery.

A Committee on the Conservation of Our Natural Resources was appointed by the Council. It consists of George F. Swain, *Chairman*, L. D. Burlingame, Charles Whiting Baker, M. L. Holman, and Calvin W. Rice.

A communication from the National Conservation Commission was received, requesting the assistance of the Society by the appoint-

ment of an Advisory Board for the purpose of valuing water power. In response to the request, the Council appointed John R. Freeman, Geo. F. Swain and Charles T. Main, a special committee of the Society to serve until the Annual Meeting of 1909, and voted that the expenses of the board be defrayed by the Society.

REPORTS OF THE STANDING COMMITTEES, 1908

The Finance Committee

On October 17, 1907, The Mechanical Engineers Library Association was consolidated with the Society under the corporate name of The American Society of Mechanical Engineers, in connection with which the surplus of the M. E. L. A., including the profits from the sale of the house at 12 West 31st Street, the whole amounting to $83 632.15, was transferred to the treasury of the Society. Of the funds made available from this source, $63 000 was expended in the reduction of the mortgage on the land on which the building of the United Engineering Societies was erected. The balance was returned to the treasury of the Society and appears in the statement below.

RE-INVESTMENT OF TRUST FUNDS

During the year the several trust funds of the Society have been reimbursed to the extent of the investments made by them in assuming a part of the obligations incurred in connection with the erection of the Engineering Societies Building. The principals of each of the several trust funds have been restored to the full amount and the unexpended income at 4 per cent has been added in each case. It will be of interest to note the history of these funds.

Library Fund. The Library fund was first established at the Council meeting held on June 25, 1884. From this time the contributions have been received for this fund and the interest used in paying for the purchase of books and the maintenance of the library.

In April, 1902, as a result of a joint meeting of the Finance and Executive Committees, a pamphlet was issued containing a financial report and certain recommendations which were subsequently adopted by the Council. One of these resolutions (See p. 9, vol. 24 of the Transactions, also Council Records) provided that one per cent of the gross receipts from membership dues during the fiscal year should be credited to the library fund. The intent of the com-

mittees appears to have been that the income of these funds be used for the purchase of new books, building and models. At the present time this fund amounts to $4902.71.

Weeks Legacy Fund. This fund was established in May, 1904, by bequest from the estate of Geo. W. Weeks, a member of the Society, who donated an amount to the Society for library purposes. The amount of the bequest was $1957.

Life Membership Fund. This fund was established in accordance with Article 22 of the Constitution. The first contribution was received May 12, 1880. At the present time there are 104 Life Members, and the fund amounts to $35151.07. The income is used for the current expenses and appears under the classification of dues.

Reserve Fund. This fund originated at a meeting of the Financial and Executive Committees in March 1902, and was established by the Council April 22, 1902. The evident purpose of this fund is that the initiation fees shall be treated as a special reserve fund and that at the end of each fiscal year ten per cent of the entire amount in the fund be transferred to the annual income. This fund at the end of this fiscal year amounts to $32 759.95.

Thurston Memorial Fund. This fund was established soon after the decease of Dr. Robert H. Thurston, the first president of the Society, and consists of subscriptions for the purpose of providing for the Society House a bronze bust of Dr. Thurston as a memorial. At the end of the fiscal year this fund amounted to $797.17.

It would seem advantageous at this time to enlarge upon the fact that the Society is in a position to receive and invest trust funds for the development of the various features of its work. Endowments for the benefit of the Society's activities, and to secure their continuance, will be highly appreciated and can be utilized most advantageously.

The details of the Society's investments and resources are shown in the statement herewith submitted.

Respectfully submitted,

Anson W. Burchard, *Chairman*
Arthur M. Waitt
Edward F. Schnuck
J. Waldo Smith
A. C. Dinkey

Finance Committee

PEIRCE, STRUSS AND COMPANY
CERTIFIED PUBLIC ACCOUNTANTS
40 Cedar St., New York

November 4, 1908

MR. ANSON W. BURCHARD
CHAIRMAN FINANCE COMMITTEE
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
29 West 39th Street, New York

Dear Sir:

Having audited the books and accounts of The American Society of Mechanical Engineers for the year ended September 30, 1908, we hereby certify that the accompanying Balance Sheet is a true exhibit of its financial condition as of September 30, 1908, and that the attached statements of Income and Expense and Cash Receipts and Disbursements are correct.

PIERCE, STRUSS AND COMPANY
Certified Public Accountants

PEIRCE, STRUSS AND COMPANY
CERTIFIED PUBLIC ACCOUNTANTS
40 Cedar St., New York

November 4, 1908

MR. ANSON W. BURCHARD
CHAIRMAN FINANCE COMMITTEE
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
29 West 39th St., New York

Dear Sir:

In accordance with your instructions, we have audited the books and accounts of The American Society of Mechanical Engineers for the year ended September 30, 1908.

The results of this examination are presented in three exhibits, attached hereto, as follows:

Exhibit A Balance Sheet, September 30, 1908.
Exhibit B Income and Expenses for year ended September 30, 1908.
Exhibit C Receipts and Disbursements for year ended September 30, 1908

We beg to present, attached hereto, our Certificate to the aforesaid exhibits

Yours very truly,

PEIRCE, STRUSS AND COMPANY
Certified Public Accountants

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
EXHIBIT A
Balance Sheet, September 30, 1908

ASSETS

Cash in bank	$13 458.98	
Petty cash on hand	250.00	$ 13 708.98
New York City 3½ per cent bonds 1954, par $35000		30 925.00
Equity in Society Building (25 to 33 West 39th St.)	353 346.62	
Equity, one-third cost of land (25 to 33 West 39th St.)	180 000.00	533 346.62
Library books	$13 727.10	
Furniture and fixtures	3 084.02	
Stores, including plates and finished publications	10 875.55	27 686.67
Accounts receivable		
Membership dues	$4 439.75	
Initiation fees	705.00	
Sale of publications, etc	5 952.26	11 097.01
Advances account of land subscription fund		7 960.94
Advanced payments		4 819.47
Total assets		$629 544.69

LIABILITIES

Funds		
Land fund subscriptions	$ 1 475.84	
Library development fund	4 902.71	
Robert H. Thurston memorial fund	797.17	
Weeks legacy fund	1 957.00	
Life membership fund	35 151.07	$44 283.79
United Engineering Society Building (for cost of land)		81 000.00
Membership dues paid in advance	326.05	
Initiation fees paid in advance	40.00	366.05
Reserve (initiation fees)		32 759.95
Appropriation available to complete Volume 29 of Transactions		4 451.31
Membership dues uncollected	4 439.75	
Initiation fees uncollected	705.00	5 144.75
Current accounts payable		1 352.21
Surplus		460 186.63
Total liabilities		$629 544.69

EXHIBIT B

Income and Expenses for the Year Ended Sept. 30, 1908

INCOME

Membership dues	$46 891.80	
Initiation fees	3 639.99	
Interest library funds	223.50	
Sales, publications, badges, etc	1 906.97	
Adjustment of cost of building	3 346.62	
Miscellaneous	1 072.65	
		$57 081.53

EXPENSES

Transactions, volume 29, including estimated cost to complete	$ 6 100.00*	
Office administration, including salaries	20 497.13	
Meetings, annual, spring and monthly	5 237.99	
Proceedings	8 015.63	
Membership development	2 029.73	
Pocket list and year book	2 622.57	
Library	2 539.85	
United Engineering Society assessments	6 700.00	
Miscellaneous	1 526.16	
Total		55 269.06
Excess of income over expenses		1 812.47
		$57 081.53

*Actual cost to date	$1648.69
Estimate of amount required to complete	4451.31
Estimate of amount required to complete	$6100.00

EXHIBIT C

Receipts and Disbursements for Year Ended Sept. 30, 1908

RECEIPTS

Membership dues and initiation fees	$51 442.43	
Membership dues and initiation fees, paid in advance	351.05	
Sales of publications, badges, advertising, etc	11 731.59	
Subscriptions to land fund	3 558.00	
Subscriptions to expense of annual meeting	1 865.00	
Transportation: Annual and Spring meeting	178.75	
Interest	1 504.56	
Cash exchanges, *per contra*	625.00	
Robert H. Thurston memorial fund	788.32	
Sale of property of Mechanical Engineers Library Association	83 632.15	
	$155 676.85	
Cash in banks and on hand, September 30, 1907	30 292.44	$185 969.29

DISBURSEMENTS

Disbursements for general purposes	$73 047.72	
Reduction of mortgage on land	63 000.00	
Interest on mortgage on land	4 278.08	
Investment in New York City bonds ($3\frac{1}{2}$ per cent 1954)	30 925.00	
Accrued interest paid on New York City bonds	384.51	
Cash exchanges, *per contra*	625.00	
	$172 260.31	
Cash in banks and on hand September 30, 1908	13 708.98	
		$185 969.29

Detail Statement Showing Receipts and Disbursements of the Trust Funds

LAND FUND

On hand October 1907	$2589.64
Received during the year	3558.00
Interest	84.86
	$6232.50
Interest on Mortgage Land Fund Committee	4756.66
Balance on hand	$1475.84

LIBRARY DEVELOPMENT FUND

On hand October 1, 1907	$1929.06
Received from surplus	2500.00
Membership fees	473.65
On hand	$4902.71

Interest $155.01 applied to current library expenses

ROBERT H. THURSTON MEMORIAL

Original fund	$788.32
Interest accrued	8.85
Total fund	$797.17

WEEKS LEGACY

On hand January 31	$1957.00

Interest $68.49 applied to purchase of library books

LIFE MEMBERSHIP

Balance October 9, 1907	$35 151.07

Interest $1109.20 applied to current expenses

The Meetings Committee

The results of the year's work of your Committee on Meetings are better shown in the records of the nine meetings which have been held (seven monthly, besides the regular annual and summer conventions) and in the regular monthly publication of the Society, than they can be in any formal report. That the meetings have been highly successful in point of attendance and in the interest displayed by the membership is well known. We believe the summer convention at Detroit was particularly notable for the large attendance, the attractive excursions and other social features, and for the interest arising from the holding at the same time in the same city of the conventions of kindred organizations: viz., the Society for the Promotion of Engineering Education, the Society of Automobile Engineers and the Gas Power Section of our own Society. The rapid multiplication of engineering societies makes coöperation between the various groups in the choice of time and place of the annual conventions exceedingly advantageous.

Your committee has continued the policy adopted some years ago and has endeavored at each semi-annual meeting to secure groups of papers bearing upon some one or more subjects of current importance to the profession. We may cite by way of illustration the symposium on the conveying of materials presented at the Detroit convention, and the session devoted to current improvements in machine shop practice which is on the program for the coming annual meeting at New York. We believe that by concentrating attention upon special subjects in this way more valuable information is elicited and is brought together in one place, which makes it much more useful for reference.

Your committee has further continued its efforts to have the papers read before the Society cover a wide variety of professional work, so that the Society's publications may be made practically useful to all its members. While much has been accomplished in the matter, much remains to be done. There are still important departments of mechanical engineering work whose processes and methods and data are not recorded in our Transactions nor in those of any other engineering Society.

Undoubtedly the most important departure of the year just closed has been the engagement of Mr. Lester G. French as Editor of the Society's publications. With the great increase in the number of papers offered to the Society, it became absolutely necessary to have

a competent editor to work in coöperation with your committee in the solicitation, examination and preparation of papers for publication. The interest of the meetings and the quality of the papers presented during the past year have been in no small degree due to Mr. French's ability and industry.

In order that this new departure in the Society's work might be carried on without burdening the Society's treasury, it was decided to insert advertising pages in the Society's Journal, and this change was inaugurated with very satisfactory results in the September number.

As a natural result of the expansion of the Society's work, the number of papers submitted to your committee for approval has very greatly increased, so that it becomes necessary to reject a considerable proportion of those which are offered. We believe this condition is one on which the Society is to be congratulated. The aim should be to make our publications of such high rank that it will be esteemed an honor for an engineer to have a paper accepted and published by our Society. This condition can only be brought about, however, if the number of papers offered continues materially in excess of the number which will be used, since it is only when such a surplus exists that proper selection becomes possible.

The objection will doubtless be raised that it is a disappointment to an individual author if after spending much time in the preparation of a paper it is finally rejected by the Society. The answer to this objection is that the interests of the profession—of the Society as a whole—are more important than the interests of any individual. If our Society finds itself obliged to reject an offered paper, its author is at liberty to seek publication through other channels.

There are, however, certain papers which by reason of great length, or because they appeal only to some small branch of the profession, would not be accepted by any journal published by private enterprise and are for the same reasons undesirable for our Society publications. We believe that certain papers of this class, where they contain material which ought in some form to be made available for record, can be treated in the way which has long been practiced by the Institution of Civil Engineers of Great Britain. That is, the paper as a whole can be filed in the Society's library for reference, and either its title only or some brief summary of the writer's conclusions and recommendations can be published in the Society's Journal.

By proper adoption of such a policy it should be possible for the Society to increase the number and variety of papers in its Trans-

actions without increasing the bulk and expense of the volume. All of which is respectfully submitted.

CHARLES WHITING BAKER, *Chairman*	
W. E. HALL	*Meetings Committee*
WM. H. BRYAN	
L. R. POMEROY	
CHARLES E. LUCKE	

THE MEMBERSHIP COMMITTEE

The Membership Committee has, during the year, considered 399 applications for membership.

It has recommended that there be placed on the

Spring Ballot	139 names
Fall Ballot	190 names
Total on Ballot	329 names
It has not recommended	68 names
Total	397 names

Of the 329 recommended for ballot, 289 were new candidates for membership and 40 were promotions.

There are 87 applications, which have come in since the list was prepared for the Fall Ballot, which have not been considered by the Committee.

The experience of the past five years under the new Constitution has brought forcibly to the attention of the Committee several questions.

The question which has given the Committee probably the greatest trouble relates to C10 of the Constitution which attempts to define the qualifications of an Associate.

C10, as written, seems to call for *three different classes of Associates.*

a Men who would be qualified to be Members, *if they were old enough.*

b Men who would be "competent to take charge of engineering work" but supposedly *not* "*responsible*" charge.

c Men who would be qualified to "coöperate with engineers."

These *three classes in one grade* lead to confusion in the minds of

the Committee and must also be confusing to the members of the Society.

It is the opinion of the Committee that C10 of the Constitution should be rewritten so as to make it more consistent and more explicit.

The Associate grade should not be a mere stepping stone from the Junior to the Member grade. The minimum age of the Associate and Member should be the same, namely 30 years. Persons should not be received into the Society as Juniors who are over 30 years of age.

The average age of Juniors on this fall's ballot is about 25.5 years. The average Junior will therefore have to wait only 4.5 years before he can apply for promotion; but during that time he will probably have decided on his career and will be able to state his experience and qualifications so that the Committee can decide intelligently whether to recommend him for Member or Associate.

It is the opinion of the Committee that C10 of the Constitution should be so worded as to include that large class of men who, while not engineers, may, by reason of their connection with science, the arts or industries, or engineering construction, contribute to the advancement of professional knowledge in engineering.

The number of Associates should be limited to 40 per cent of the total voting membership.

The Committee recommends the following, to take the place of the present C10, of the Constitution, as embodying its views.

> "C10 An Associate shall be 30 years of age or over. He must have been so connected with some branch of Engineering, or Science, or the Arts, or Industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge. He need not be an engineer."

The Committee also recommends the following to be added at the end of C11 of the Constitution.

> "A person who is over 30 years of age cannot enter the Society as a Junior."

The Committee particularly calls the attention of members of the Society who are supporting applicants for membership, to the desirability of giving information about such applicants as is *within their personal knowledge. "Hearsay" information is of very little value*

to the Committee in determining the fitness of an applicant for membership.

Respectfully submitted

JESSE M. SMITH, *Chairman*
HENRY D. HIBBARD
C. R. RICHARDS
FRANCIS H. STILLMAN
GEO. J. FORAN
Membership Committee

THE PUBLICATION COMMITTEE

The Committee has examined carefully all of the papers presented at the meetings of the Society with a view of passing upon their suitability for publication in the Transactions. In this work it has endeavored to maintain the dignity of the Transactions, and to this end has considered it its duty to omit any objectionable personalities that might be found in the papers or in the discussions of the same. It has also endeavored to omit portions of papers or discussions where this could be done without eliminating ideas or data.

Proceeding in accordance with the above plan, the Committee has found it advisable to omit the publication of several papers, and in other cases has eliminated a considerable amount of matter from the text.

COMBINED SUMMARY OF COST OF VOLUMES

Volume number	Year	Total cost	Cost omitting advance papers and revises	Advance papers	Revised papers	Number of copies	Number of pages	Total cost per copy	Total cost per 100 pages
19	1898	$10,973	....	....	....	2250	1033	$4.87	$0.471
20	1899	9,643	....	....	....	2300	1035	4.19	0.408
21	1900	12,798	....	....	....	2500	1177	5.12	0.435
22	1901	12,740	....	....	....	2600	1164	4.90	0.421
23	1902	10,678	$8,730	$1,249	$699	2600	878	4.10	0.467
24	1903	15,479	12,080	1,301	2,098	2700	1563	5.62	0.359
25	1904	15,146	11,006	2,864	1,276	3000	1155	5.04	0.436
26	1905	11,830	8,417	2,650	763	3300	841	3.58	0.425
27	1906	14,546	10,127	2,678	1,741	3750	999	4.14	0.414
28	1907*	18,930	9,925	....	....	4000	981	4.73	0.482

*$18 930 includes cost of Proceedings from October, 1, 1906, to September 30, 1907, also the cost of Transactions from October 1, 1906, to December 31, 1906. Previous to the publication of Volume 23, no separate accounts were kept for Transactions, the same being included in general expense for printing.

The Committee has made no endeavor to check up the detailed expenditures involved in publishing the Transactions, but has had the Secretary tabulate the expenditures involved in printing the Transactions for the last and previous years. The results of the tabulation are shown in the accompanying table.

The costs given in the table include the amount expended in bringing out advance papers before the Proceedings took their place, so that the figures given include the expense of all the printing charges in preparing the Transactions. The costs for Vol. 28 are in some cases estimated, whereas all the figures for other years represent the amount actually expended. It may be noted that the total cost for Vol. 28 is higher than for the preceding years. This is accounted for by the large number of folders in this volume, and by the fact that additional expense items are included to cover certain editorial work made necessary through a change in the form of the Proceedings.

Respectfully submitted

D. S. JACOBUS, *Chairman*
FRED J. MILLER
WALTER B. SNOW
H. W. SPANGLER
H. F. J. PORTER

Publication Committee

THE LIBRARY COMMITTEE

During the past year the conduct of the Library has been modified to the extent that the administration has been placed in the hands of a Chief Librarian in common with the libraries of the American Institute of Mining Engineers and the American Institute of Electrical Engineers. All employees are paid by the United Engineering Society, this expense being distributed among the Founder Societies. This action is in accordance with the action of the Council and the Detroit meeting, pursuant to recommendation made by the Chairman of the Library Committee after conference with the Chairman of the Founder Societies and the Trustees of the United Engineering Society. At the Detroit meeting the Council adopted the following resolutions:

First: That the administration of the library be placed in the hands of the chief librarian, all employees of the library to be subject to the direction of the

said chief librarian. It was agreed that Miss L. E. Howard should be appointed chief librarian.

Second: That each new employee in the library shall be approved both by the authorized representative of the Founder Societies interested, and by the chief librarian.

Third: In the event of any employee not being satisfactory to said representative of the Founder Societies interested, or to the chief librarian, the fact shall be reported by the chief librarian to the House Committee, which shall have the power to act.

Fourth: That in the absence of the chief librarian, the House Committee shall have the right to specify an employee of the library to act as chief librarian, during such period of absence.

Fifth: That the United Engineering Societies be requested to pay the salaries of all the library employees, and the amount to be repaid by the Founder Societies respectively shall be distributed in such proportion as may be jointly agreed upon.

This action relates solely to the administration of the Library, the acquisition and ownership of the books remaining wholly with the Society.

A gratifying number of books has been acquired by purchase during the year.

The Society has received from Prof. John E. Sweet, Past-President and Honorary Member of the Society, an original copper-plate impression of the Certificate of Membership in the Insurance Society of the Steam Engine Works of Messrs. Boulton & Watt at Birmingham also a photograph of Watts' workroom at Heathfield Hall where this certificate was discovered. Mr. H. H. Suplee, Member of the Society, has presented a photograph of the original letter of Robert Fulton offering the steamboat to the French government in 1803, together with a photograph of the drawing which accompanied it.

Mr. Henry R. Towne, Past-President of the Society, and Chairman of the Mechanical Engineers Library Association presented 150 books to the Library. Many of these books were originally in the Library of Mr. Towne's father, John H. Towne, and are especially interesting as early works on engineering. A complete list was published in the Proceedings of January, 1908.

Mrs. Annie M. Forney presented to the Library the collection of engineering books which belonged to her husband, the late Matthias Nace Forney. The collection includes 158 books on engineering subjects, 19 handbooks, many volumes of Transactions of engineering societies and bound volumes of periodicals; also portraits of George Westinghouse, Charles R. Johnson and Howard Fay, and a photograph of a Baldwin locomotive. A complete list of the books was published in the April 1908 Proceedings.

Henry Harrison Suplee, Chairman of the Library Committee of this Society, gave to the library a collection of sixty books comprising miscellaneous works and a full set (36 volumes) of the International Library of Technology. A complete list was published in the November 1908 Journal.

It is to be hoped that other members of the Society will imitate these examples and make the Library the place for the deposit and preservation of books and engineering relics of value and interest. Gifts are especially desired of back numbers and volumes of periodicals to complete the sets now in the library, and an examination of the printed list of the sets now in the library will enable the gaps to be ascertained. Technical and engineering books are always welcome and any communications addressed to the Secretary of the Society upon these matters will be referred to the proper individuals for attention.

At the present time the Library receives 163 periodicals in exchange for its publications, and 40 periodicals as gifts.

The members of the Founder Societies should impress upon themselves the large opportunity for information and research afforded by the Library of the Engineering Societies.

It has a collection of 50 000 volumes of scientific and engineering works, and 450 current technical journals and magazines of Europe and America are readily accessible on the files of the reading room. The library is open every day except Sundays and holidays from 9 a.m. to 9 p.m. and librarians are constantly in attendance who will assist in finding information which the members, and users of the library may not be able to locate.

A feature which is of continuous service is a file of trade catalogues. An index to this file, both by subject and firm name, is kept, to make the information easily accessible.

Members are requested to recommend the purchase of books not in the library which in their opinion would add to its reliable resources. Cards are kept at the librarian's desk for that purpose and such recommendations to purchase are immediately transmitted to the Library Committee for consideration.

Members should invite their friends in the profession to make use of the library. Such a storehouse of information should be largely consulted and freely used for the advancement of science and engineering. The fact that it is a reference library free to all should be generally advertised.

The equipment and facilities of the library, located as it is in the

headquarters of engineering—in a fireproof building with every convenience for indexing and caring for its gifts—must recommend it to friends who wish to place scientific and engineering literature or valuable engineering relics where they will be of the widest enduring influence.

It is maintained as a free reference library and with the idea of being of the greatest use and benefit to the profession, regardless of society affiliations. Librarians are always in attendance to give assistance in locating information, and it is the expressed aim of the Engineering Societies to make it of permanent and ever-increasing service.

At the present time the Library Development Fund amounts to $4902.71 and the income for the year just ended was $155.01. The Weeks legacy is $1957.00, yielding an income of $68.49 for the fiscal year. These items of income have been used for the purchase of new books.

The Committee desire to place on record their appreciation of the work of Miss Isabel Thornton, the Librarian of the Society for the past thirteen years, who resigned her connection with the Library in October last.

At a meeting of the Council of the Society November 10, a vote of thanks to Miss Thornton for her long and valued services was passed, and she carries with her the best wishes of all the members of the Society.

Respectfully submitted

H. H. Suplee, *Chairman*
A. W. Howe
Ambrose Swasey
Leonard Waldo
G. M. Basford

Library Committee

REPORTS OF SPECIAL COMMITTEES, 1908

The Society's Land Problem

At the annual meeting, December 1907, in New York, there was presented the report of the special committee appointed to raise funds to pay the Society's share of the cost of the land on which the new building stands.

By this report, which was placed before all the membership in the Proceedings for December (p. 529), it is seen that of the total

amount so far raised $55 500 has come from manufacturing concerns who were asked to contribute on the ground of their interest in mechanical engineering and the work of the Society; the members having subscribed up to that time a total of $15 500.

The complete list of subscriptions received from manufacturing and other firms, as it now stands, is contained in the appendix to this report.

As stated in the report referred to above, the present is not a favorable time for subscriptions from manufacturing firms. We shall probably secure a few more of such subscriptions but it would seem as though the membership ought to do much more than has so far been done. As it now stands only about 7 per cent of the membership has subscribed. These are listed in the appendix.[1]

Of those members who have not subscribed, a number have written the committee, expressing sympathy with the work but regretting their inability to subscribe. This leaves a very large proportion of the membership from which no reply has been received.

It would seem that every member of the Society ought to be sufficiently interested in the Society's welfare to make response to such an appeal in its behalf.

The number of members who have not subscribed being about 2900, and there being yet to raise $73 600, if those members who have not yet subscribed will subscribe an average of $25.38 each, the debt will be lifted from the Society and it will be free to devote its income entirely to lines of activity already inaugurated or in process of planning, which will enhance the value of membership, especially for those who live at a distance from the headquarters.

A subscription does not necessarily imply immediate payment but only a promise to pay at such time or times as may be convenient for the subscriber.

It is earnestly hoped that every member who reads this will give it his most careful attention; that he will not merely send in his own subscription, but will speak to other members about it and urge them to subscribe as liberally as possible.

There are undoubtedly a few members, who for one perfectly valid reason or another, cannot subscribe the amount mentioned. The deficiency must be made up by others who are fortunately able to subscribe more. If every member does what he is able to do the amount needed will be quickly raised.

[1] The Society by vote at the annual meeting decided that it would **not be** advisable to publish the amounts of individual subscriptions.

We earnestly ask every member to do what he can and to do it immediately.

FRED J. MILLER, *Chairman* JAMES M. DODGE ROBERT C. MCKINNEY	*Land and Building Fund Committee*

SUBSCRIPTIONS FROM FIRMS AND CORPORATIONS

Allis-Chalmers Co., Milwaukee, Wis	$1166.67
Almond Manufacturing Co., T. R., Brooklyn, N. Y	100.00
American Locomotive Co., New York	7500.00
American Pulley Co., Philadelphia, Pa	300.00
Babcock & Wilcox Co., The, New York	3000.00
Baird Machinery Co., Pittsburg, Pa	25.00
Bettendorf Axle Co., Davenport, Iowa	100.00
Beckford Drill and Tool Co., Cincinnati, O	100.00
Billings & Spencer Co., The, Hartford, Conn	100.00
Bradford Machine Tool Co., Cincinnati, O	50.00
Bridgeport Brass Co., Bridgeport, Conn	25.00
Brown & Sharpe Manufacturing Co., Providence, R. I	1000.00
Brownell Co., Dayton, O	50.00
Buckeye Engine Co., Salem, O	100.00
Bullard Machine Tool Co., Bridgeport, Conn	100.00
Burroughs Adding Machine Co., Detroit, Mich	100.00
Cadillac Motor Car Co., Detroit, Mich	50.00
Cincinnati Machine Tool Co., Cincinnati, O	25.00
Cincinnati Milling Machine Co., Cincinnati, O	250.00
Cincinnati Planer Co., Cincinnati, O	50.00
Colt's Patent Fire Arms Manufacturing Co., Hartford, Conn	100.00
Continental Iron Works, Brooklyn, N. Y	1000.00
Dreses Machine Tool Co., Cincinnati, O	50.00
Ellsworth & Co., J. W	3.33
Engineering Magazine, New York	300.00
Engineering News, New York	250.00
Engineering Record, New York	1000.00
Ferracute Machine Co., Bridgeton, N. J	100.00
Fletcher Co., W. & A., Hoboken, N. J	100.00
Gisholt Machine Co., Madison, Wis	200.00
Gleason Works, Rochester, N. Y	100.00
Hayes Manufacturing Co., Detroit, Mich	25.00
Heine Safety Boiler Co., St. Louis, Mo	300.00
Hess-Bright Manufacturing Co., Philadelphia, Pa	500.00
Hill Publishing Co., New York	2500.00
Houston Stanwood & Gamble Co., Cincinnati, O	100.00
Industrial Press, The, New York	500.00
Jenkins Bros., New York	250.00
Kennedy Valve Manufacturing Co., The, Coxsackie, N. Y	50.00

LeBlond Machine Tool Co., R. K., Cincinnati, O.	$100.00
Lidgerwood Manufacturing Co., New York.	500.00
Link-Belt Co., Philadelphia, Pa.	2000.00
Littleford Bros.,	25.00
Locomobile Co. of America, The., Bridgeport, Conn	100.00
Lodge & Shipley Machine Tool Co., Cincinnati, O.	200.00
Lucas Machine Tool Co., Cleveland, O.	50.00
Lunkenheimer Co., Cincinnati, O.	250.00
McIntosh Seymour & Co., Auburn, N. Y.	100.00
Machold & Riddell, Philadelphia, Pa.	50.00
Messnick Manufacturing Co., Detroit, Mich.	25.00
Moore & Co., Chas. C., San Francisco, Cal.	250.00
Niles-Bement-Pond Co., New York.	5000.00
Quintard Iron Works Co., New York.	100.00
Roe Stephens Manufacturing Co., Detroit, Mich.	25.00
Sellers & Co., Inc., Wm., Philadelphia, Pa.	1000.00
Solvay Process Co., Syracuse, N. Y.	5000.00
Triumph Electric Co., Cincinnati, O.	50.00
Underwood Typewriter Co., Hartford, Conn.	100.00
Union Twist Drill Co., Athol, Mass.	100.00
United Engineering & Fdy. Co., Pittsburg, Pa.	500.00
Ward's Engine Works, Charleston, W. Va.	150.00
Watson-Stillman Co., New York.	500.00
Westinghouse Elec. & Manufacturing Co., Pittsburg, Pa.	16 667.67
Whitney Manufacturing Co., Hartford, Conn.	50.00
Wiley & Sons, John, New York.	100.00

OTHER CONTRIBUTORS

Abbott, W. L., Chicago, Ill.
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Andrew, James D., Boston, Mass.
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Baldwin, Stephen W., Brookline, Mass.
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Barnes, S. G., Detroit, Mich.
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Langer, P., Aachen, Germany
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Norris, R. V., Wilkesbarre, Pa.
Otis, Spencer, Chicago, Ill.
Park, W. R., Boston, Mass.
Parkhurst, F. A., Bridgeton, N. J.
Pettis, C. D., Chicago, Ill.
Philbrick, Frank B., Waterville, Me.
Platt, John, New York
Polson, J. A., E. Lansing, Mich.
Power, Fred M., Solvay, N. Y.
Prather, H. B., Cleveland, O.
Prescott, F. M., Milwaukee, Wis.
Price, A. M., Elgin, Ill.
Quimby, W. E., New York
Quint, A. D., Hartford, Conn.
Reed, E. Howard, Worcester, Mass.
Reid, Marcellus, Cleveland, O.
Reiss, Geo. T., Hamilton, O.
Rhoads, G. E., Altoona, Pa.
Rice, Calvin W., New York
Richter, Ernst, Cincinnati, O.
Rider, Geo. S., Cleveland, O.
Riggs, John D., Chicago, Ill.
Robeson, A.M., Johannesburg, S. Africa
Roper, Norman B., Cudahy, Wis.
Sargent, Fredk., Chicago, Ill.
Satherberg, Carl H., Philadelphia, Pa.
Saunders, W. L., New York
Schaeffler, Joseph C., Boston, Mass.
Scheele, John M. B., New York
Scheffler, Fred. A., Glen Ridge, N. J.
Schneider, Paul E., Garfield, N. J.
Schulte, G. H., Milwaukee, Wis.
Schwarz, F. H., Lawrence, Mass.
Smith, E. J., Allentown, Pa.
Smith, Jesse M., New York
Smith, Oberlin, Bridgeton, N. J.
Smith, O. G., Dayton, O.
Snow, Wm. G., Boston, Mass.
Snow, Wm. W., Hillburn, N. Y.
Sprado, Ralph, Chicago, Ill.
Stanwood, J. B., Cincinnati, O.
Starbuck, G. F., New Haven, Conn.
Stetson, G. R., New Bedford, Mass.
Stevens, John A., Lowell, Mass.
Sulzer, J. Carl, Winterthur, Switzerland
Swasey, Ambrose, Cleveland, O.
Sweet, John E., Syracuse, N. Y.
Symonds, Nathaniel G., Indianapolis, Ind.
Terry, C. D., Kewanee, Ill.
Thomas, Charles W., New York
Thompson, A. W., Manchester, N. H.
Thomson, John, New York
Tompkins, Stonewall, Miller School, Va.
Torrance, Henry, Jr., New York
Tower, D. W., Grand Rapids, Mich.
Towle, Wm. M., Enosburg Falls, Vt.
Towne, Henry R., New York
Trump, Edward N., Syracuse, N. Y.
Tufts, Leonard, Pinehurst, N. C.
Turner, John, E. Orange, N. J.
Tuttle, W. B., San Antonio, Texas
Tyler, Chas. C., Ilion, N. Y.
Upson, Maxwell, New York
Van Nostrand, L. G., Scranton, Pa.
Vauclain, S. M., Philadelphia, Pa.
Vaughan, H. H., Montreal, Canada
Veeder, C. H., Hartford, Conn.
Viola, B., Brooklyn, N. Y.
Waitt, A. M., New York
Wallace, D. A., Newark, N. J.
Ward, Charles E., Charleston, W. Va.
Warner, W. R., Cleveland, O.
Warren, John E., Cumberland Mills, Me.

Wehner, Lewis, Milwaukee, Wis.
Wellman, S. T., Cleveland, O.
Weymouth, C. R., San Francisco, Cal.
Wheeler, C. H., Philadelphia, Pa.
Whitaker, H. E., Detroit, Mich.
White, John C., Madison, Wis.
Whitehead, George E., Providence, R.I.
Whyte, John S., Glace Bay, Cape Breton, N. S.
Wick, Henry, Youngstown, O.
Wildin, G. W., New Haven, Conn.
Willcox, Charles H., Westport, Conn.
Williamson, George E., Bridgeport, Conn.
Willis, E. J., Richmond, Va.
Wolf, Fredk. W., Chicago, Ill.
Woodbury, C. J. H., Lynn, Mass.
Young, John P., Youngstown, O.

APPENDIX TO THE REPORTS

MEMORIAL EXERCISES FOR LORD KELVIN

The memorial exercises in honor of Lord Kelvin, Hon. Mem. A. I. E. E., were held under the auspices of the American Institute of Electrical Engineers, on January 12. This Society participated and was officially represented by Rear-Admiral Melville, Andrew Carnegie, B. F. Isherwood, George Westinghouse and Thomas A. Edison.

Rear-Admiral Melville, who was one of the speakers, gave an account of Lord Kelvin's work in relation to the Navy. Other speakers were Dr. Manning, of Trinity Church, and Professor Elihu Thomson, who spoke of Lord Kelvin's inventions of measuring instruments, his participation in the determination of the C. G. S. system, and in the electrical development of Niagara.

Mr. George G. Ward, Local Hon. Sec. Institution of Electrical Engineers, Great Britain, mentioned Lord Kelvin's part in the laying of the first Atlantic cable, and his invention of the syphon recorder.

Mr. T. C. Martin outlined Lord Kelvin's interest in the formation of the American Institute of Electrical Engineers, and Professor E. L. Nichols, President of the American Association for the Advancement of Science, gave a most complete address on the work of Lord Kelvin in all branches of science, and pronounced him the greatest man of science in this generation.

PARTICIPATION OF THE ENGINEERING SOCIETIES IN THE CONFERENCE ON THE CONSERVATION OF OUR NATURAL RESOURCES

At the conference on the Conservation of Natural Resources held at the White House in Washington, May 13–15, in response to the invitation of the President of the United States, there were present the Governors and their appointed delegates, members of the Cabinet and Supreme Court, Senators, Congressmen and representatives of the leading engineering, scientific and educational societies of the country.

The four national engineering societies were represented by their presidents and secretaries, respectively: Charles Macdonald, President, and Charles Warren Hunt, Secretary, of the American Society of Civil Engineers; John Hays Hammond, President, and Dr. Joseph Struthers, Assistant Secretary, of the American Institute of Mining Engineers; M. L. Holman, President, and Calvin W. Rice, Secretary, of The American Society of Mechanical Engineers; and Henry G. Stott, President, and Ralph W. Pope, Secretary, of the American Institute of Electrical Engineers.

The engineering and technical press were represented by Charles Whiting Baker, Editor, *Engineering News*; Charles Kirchhoff, Editor, *Iron Age*; T. C. Martin, Editor, *Electrical World*, and Walter R. Ingalls, Editor, *Engineering and Mining Journal*.

The engineering profession was further represented by the following delegates, chosen either by the Governors, or as representatives of organizations.

Otto von Geldern, California; Thomas W. Jaycox, Colorado; Isham Randolph, Illinois; Lyman E. Cooley, Illinois; John B. Atkinson, Kentucky; Col. Jas. A. Ockerson, Missouri; J. E. Sirrine, South Carolina: Mem. Am. Soc. C. E.

Paul A. Fusz, Montana: Mem. Am. Inst. M. E.

Prof. Geo. F. Swain, Massachusetts; Andrew Carnegie, New York: Hon. Mem. Am. Soc. M. E.

John H. Finney, Georgia; Alonzo Gartley, Hawaii; J. W. Porter, Kentucky; W. C. Barnes, New Mexico, H. S. Putnam: Mem. Am. Inst. E. E.

The total representation by members of the four national engineering societies was 25.

On the evening of May 12, previous to the convening of the delegates at the conference, a dinner was tendered to the presidents of the national engineering societies and representatives to the Congress by the members of the Washington Society of Engineers, in coöperation with other engineers of Washington. This dinner was attended by a large number of prominent engineers and several notable speeches were made indicating the interest of the engineers in the conservation problem and their desire to coöperate in every possible way for the preservation of the country's resources. This Society was represented at the dinner by President M. L. Holman and Secretary Calvin W. Rice. In his speech President Holman said:

> The engineering profession has been honored by invitations to the four national societies to be represented at this conference, and the presidents of these societies have been invited to participate. The event is one which will be recorded in history, and the result of the conference will be the guideboard marking the fork of the road, where the nation takes the road leading to prosperity or continues, like the prodigal son, to waste its inheritance in riotous living.

CONFERENCE AT THE WHITE HOUSE

The conference met on the morning of May 13 in the East Room of the White House. The audience of about 400, representative of every section of the country and of almost every professional and

industrial activity, was one of the most notable and distinguished ever assembled.

It is inevitable, also, that it will mark one of the great movements for the welfare of the human race.

The opening address by the President sounded a new note in national affairs: an appeal for the perpetuity of the nation, the welfare of posterity and devotion to public interest by the present generation. He said:

The time has come to inquire seriously what will happen when our forests are gone, when the coal, the iron, the oil and the gas are exhausted, when the soils shall have been still further impoverished and washed into the streams, polluting the rivers, denuding the fields, and obstructing navigation. Any right thinking father earnestly desires and strives to leave his son both an untarnished name and a reasonable equipment for the struggle of life. So this nation as a whole should earnestly desire and strive to leave to the next generation the national honor unstained and the national resources unexhausted.

The first session was devoted to the Mineral Resources of the country and addresses were made by Andrew Carnegie, Hon. Mem. Am. Soc. M. E., on "The Conservation of Ores and Related Minerals;" and by Dr. I. C. White on "The Waste of our Fuel Resources." The general discussion was opened by John Mitchell.

The second and third sessions were given to the subject of Land Resources, and addresses were made, by James J. Hill on "The Natural Wealth of the Land and its Conservation;" by Prof. T. C. Chamberlain on "Soil Wastage;" by R. A. Long on "Forest Conservation;" by ex-Governor George C. Pardee on "Resources Related to Irrigation;" and by Hon. H. A. Jastro on "Grazing and Stock Raising." The general discussion was opened by ex-Senator Carey.

At the fourth session Water Resources were discussed and formal opening statements were made by Dr. George M. Kober on "Conservation of Life and Health by Improved Water Supply;" by Professor Emory R. Johnson on "Navigation Resources of American Waterways; by H. S. Putnam, Mem. A. I. E. E., on "Conservation of Power Resources." These addresses were followed by a general discussion.

The final session was devoted to general discussion of the subject of the Conservation of our Natural Resources.

Mr. Carnegie, Honorary Member of the Society, made an address of which we give a short résumé:

From 1820, the date of the beginning of active coal mining in the U. S. until 1895, 4 000 000 000 tons were mined by methods so wasteful that some 600 000-

000 tons were either destroyed or allowed to remain in the ground forever inaccessible. Between 1896 and 1906, as much coal was taken from the earth as during the preceding 75 years and more than 3 000 000 000 tons were destroyed or left in the ground beyond reach of future use. To this date the actual consumption of coal has been over 7 500 000 000 tons; the waste and destruction in the neighborhood of 9 000 000 000 tons.

The rate of the consumption of coal doubles during each decade, and unless there be careful husbanding, or revolutionizing inventions, or some industrial revolution, the greater part of the estimated 2 000 000 000 000 tons of coal of our original heritage will be gone two centuries hence. And the methods of consuming coal are still more wasteful than the processes of mining. Not more than 5 to 10 per cent of the potential energy is actually used; the remaining 90 to 95 per cent is absorbed in rendering the smaller fraction available in actual work. In electric heating and lighting plants, it is much more; hardly one-fifth of one per cent or one five-hundredth part of the energy of the coal is actually utilized. These wastes are constantly decreasing through the development of gas-producers, internal combustion engines and steam turbines, but not so rapidly as to affect seriously the estimates of increase in coal consumption.

He touched upon the production of iron, copper, zinc, lead and silver, stating that as the rate of production of iron doubles each decade, by 1938 about half of the original supply of iron ore will be gone, and all the ore now deemed workable will be used long before the end of the present century.

Mr. Carnegie suggested as a remedy the lessening of the demands upon the products. One way of doing this is to substitute water-carriage for rail-carriage. This would reduce the consumption of iron by three-fourths to seven-eighths in the transportation department and at the same time reduce the consumption of coal for motive power 50 to 75 per cent, with a corresponding reduction for smelting.

The use of concrete, simple and reinforced, is decreasing the demand for iron for structural purposes, and the internal combustion engines and gas producers double or triple the power per unit of coal, and permit the use of lignite, culm, slack and inferior coals. The most promising check on coal consumption is the substitution of other power: electrical transmission of water power will doubtless soon affect the increasing drain on our coal.

In the general discussion which followed the reading of the papers, Mr. Calvin W. Rice, Secretary Am. Soc. M. E., offered the following:

Mr. President:

The Governor of Indiana has asked a question which I think should be very definitely answered, as it is probable there are other gentlemen who are also impressed with the difficulties expressed by him.

For instance, the Governor asks how one may conserve coal, and if it is proposed to stop the mining of coal. I answer, absolutely no. The American people are especially quick to take advantage of superior methods of operation and it is only necessary to point out to the progressive mine operators that it is possible, by improved methods of mining, to secure a greater percentage of the coal than they are now mining. For instance, the Governor of Kentucky has stated that the average number of deaths from accident in the mines of one of the large opera-

ting companies of his State is only one death per million tons of coal as compared with six deaths per million tons of coal on the average in the United States. In order to secure these better conditions throughout the country, therefore, it is only necessary to assist the companies in other states to approach the conditions under which the company in Kentucky is operating. That is the direct benefit of this conference. This method may be followed in the conservation of all our natural resources.

This whole subject divides itself naturally into five parts: first, inventory of our resources; second, discussion of the problem; third, statement of remedies; fourth, education of the people; fifth, legislation.

The spirit of all investigation, statement and legislation should be *constructive* rather than *prohibitive*. That is, instead of demanding that there shall be no coal mined, show how coal can be mined to better advantage; show how to design plants which shall effectively use low grade fuels, thus making a market for the coal now left in the earth, benefiting alike the operator, the miner, and the user of coal. This is the typical answer for all the problems of this kind.

In order to take up the above five steps, I recommend that each Governor immediately appoint a commission composed of a representative citizen from each of the great professions, legal, medical, and engineering, for securing information, holding hearings and promoting discussion, and reporting recommendations to the Governor. The education of the people can take place through the several channels available to the Governor, as the public press, associations, the legislature, or otherwise.

The legislation and general benefit to the people will follow as a matter of course after this complete statement before the public.

Resolutions were offered to the conference by the presidents of the four national engineering societies, representing approximately 20 000 American engineers, as follows:

WASHINGTON, D. C., MAY, 1908.

The undersigned, representing approximately 20 000 American engineers, respectfully recommend the following resolutions for adoption by this Conference:

CHARLES MACDONALD,
President, American Society of Civil Engineers.
JOHN HAYS HAMMOND,
President, American Institute of Mining Engineers.
MINARD LAFEVER HOLMAN,
President, American Society of Mechanical Engineers.
HENRY GORDON STOTT,
President, American Institute of Electrical Engineers.

Resolved: That this Conference places on record its conviction that to conserve and protect from waste and destruction the natural wealth of the United States in mines, forests, lands, and waters is of vital necessity to the public welfare. Action in this matter has been too long delayed, and vast loss has resulted in consequence, notably in the destruction wrought by forest fires, by floods, and ruin of lands whose fertility and crop-bearing power has been lost. This

unfortunate destruction of part of the natural wealth with which this virgin continent was originally stored makes it all the more necessary that wise action be taken to check further loss.

2 Though it recognizes the imperative need for prompt action, this Conference is impressed with the difficulty of framing legislative acts which shall result in the largest measure of public benefit. The problems presented are many of them new and unprecedented. It is probable that action by both the Federal Government and the individual States will be essential, and it may also be possible by suitable laws to enlist the aid of private enterprise. But to decide upon the proper distribution of responsibilty, and to frame laws which shall not work injury as well as benefit, are matters demanding most careful study and investigation by men of high standing and expert qualifications.

3 While certain individual measures may be already in such shape that action upon them may wisely be taken, this Conference holds that for the guidance of legislators, both State and Federal, a thorough investigation and study should be made by National and State Commissions so constituted that their conclusions and recommendations will be everywhere recognized as authoritative and made solely in the public interest.

4 This Conference, therefore, urges upon Congress and the State Legislatures the enactment of laws authorizing the President and the Governors, respectively, to create National and State Commissions to investigate and report upon what measures should be taken to conserve the National and State natural resources.

5 These commissions should report at the earliest possible date consistent with the thorough performance of their work, in order to enable the President and the Governors to transmit with recommendations their reports to Congress and the State Legislatures for such action as may seem advisable to protect our natural resources from further spoliation and destruction, and to secure such economy in their use as will preserve for coming generations the foundations of prosperity.

6 In order to insure the harmonious coöperation of all the Commissions, this Conference requests the President to call another National Conference at such time as may seem most advisable.

7 To secure the most efficient organization for handling the National problem which the reports of these Commissions will inevitably raise, this Conference recommends for the consideration of the President and Congress the formation of a Department of Public Works to which these and other engineering matters could be referred and to which the State Commissions could apply for information and assistance.

The resolutions that were adopted by the conference were broad in their scope, high in their ideals, and in full agreement with the call to future prosperity issued by the President and the notes of warning sounded by those responding. They advocated commissions, State and National, for the conservation of natural resources, and laws for the protection of forests; and ended with the strong recommendation for "the enactment of laws looking to the prevention of waste in the mining and the extraction of coal, oil, gas, etc., with a view to their wise conservation for the use of the people and to the protection of

human life in the mines," closing with the sentiment: "Let us conserve the foundations of our prosperity."

The Society held a meeting on the conservation of our natural resources in April. A condensed report of this meeting is published in this volume.

AMENDMENTS TO THE BY-LAWS AND RULES

By-laws

AMENDMENTS TO BY-LAWS 44 AND 45

The amendments to By-Laws 44 and 45 proposed at a meeting of the Council June 23, were confirmed November 10. The amendments are as follows:

B44 Omit the first word "that" and change "amended and annulled" to "amended or annulled."

B44 (That) Standards for the conduct of the business affairs of the Society, of its professional or business meetings and of its Committees and their activities may be established, amended or annulled by a $\frac{2}{3}$ vote of the members of the Council present at a meeting, provided that a written notice of the proposed addition or change may have been given at a previous meeting of the Council, and provided further that the Secretary shall have sent to each member of the Executive Committee (acting as a Committee on Standards) a draft of the proposed addition or change at least two weeks prior to the meeting at which they are to be voted on.

B45 Omit the first word "That."

B45 (That) Directions for the conduct of the business affairs of the Society may be established by the Secretary and the work covered shall be carried out as provided by these Directions. These Directions may be added to, amended or annulled by the Secretary, but it shall be his duty to send to each member of the Executive Committee (acting as a Committee on Standards) a draft of the change before it is put into effect.

BY-LAWS 46 TO 49

At a meeting of the Council in May 1906, the suggestion to form a Research Committee was made. It was necessary to amend C 45 of the Constitution before the Committee could be created. The amendment was passed by letter ballot closing March 4, 1907. The report of the committee on rules to govern the Research Committee

was presented to the Council February 11, 1908, and confirmed March 10, 1908.

B 46 The Research Committee shall consist of five Members, Associates or Juniors. The term of office of one member of the committee shall expire at the end of each annual meeting.

B 47 This committee shall have supervision of such research or investigations as may be directed or approved by the Council; shall correspond and collaborate with committees of kindred technical, scientific or other societies; shall keep in touch with researches conducted in other countries, which are of value to the engineer, and shall report the same quarterly to the Council.

B 48 The committee will be expected to maintain a system of announcement of results of research, and the trend of investigations, in any field, which will be of value to the engineering profession.

B 49 Gifts or bequests to the Society for the conduct of research or investigation shall be expended under the direction of the Council and shall be kept separate from other Society funds.

Rules

RULES 19 TO 26 REGARDING PROFESSIONAL SECTIONS

A request to form a Gas Power Section *affiliated* with the Society was presented at the Annual Meeting of 1907. The request was presented to the Council at its meeting December 6, 1907. The Council referred it to the Committee on Affiliated Societies which reported on January 14, 1908, suggesting rules for the government of professional sections. The rules were approved and confirmed by the Council at that meeting, and are as follows:

R 19 A professional section of the Society shall consist of Honorary Members, Members, Associates, and Juniors of The American Society of Mechanical Engineers and of other persons to be designated Affiliates as hereinafter described.

R 20 A professional section of The American Society of Mechanical Engineers may, with the approval of the Council, be organized for the consideration of any engineering, scientific, or professional topic, provided that a number satisfactory to the Council, of members of The American Society of Mechanical Engineers, unite in making written request for such an organization. Such section shall be designated as —— Section of The American Society of Mechanical Engineers—the blank being filled by the topic specialized.

R 21 The provisions of the Constitution, By-Laws, and Rules of

The American Society of Mechanical Engineers, and the precedents of the Society with respect to professional sessions for the discussion of papers shall cover the procedure of the professional sections, except that no meeting of a section shall be considered a meeting of the Society as a whole.

R 22 For the convenient conduct of its professional affairs, the section shall organize an Executive Committee of five members of the Society, under the general direction of the Council. Such officers as the section shall require, must be selected from the membership of the Society. Other committees of the section shall be appointed by its Executive Committee.

R 23 The Executive Committee of the section, subject to the approval and direction of the Secretary of the Society, shall designate a Secretary of the section, whose duties shall be those usually attaching to the Secretary of a professional session and who shall also see that the discussions of papers are satisfactorily reported and transmitted to the Secretary of the Society.

R 24 Expenditures for the purposes of the section chargeable to the Society shall be authorized by the Secretary of the Society before they are incurred, and must be provided for in the estimate and budget of the Committee on Meetings. No liability otherwise incurred shall be binding on the Society. Any expenditure not so provided shall be met by the section itself.

R 25 Engineers and others not members of the Society, but desiring to participate in meetings of the section, may enroll themselves as affiliates as heretofore provided with the approval of the Executive Committee of the section. Such affiliates shall have the privilege of presenting papers and taking part in the discussions. They shall pay $5 per annum which shall be due and payable, in advance, on October 1 of each year of their enrollment, and shall thereby be entitled to receive the Proceedings of the Society as they are issued month by month, for a period covered by their dues.

R 26 The Council of The American Society of Mechanical Engineers may, at 60 days notice, suspend or disband any section.

RULES 27 TO 34 REGARDING STUDENT BRANCHES

Requests for the formation of student branches were received from Stevens Institute of Technology and Cornell University. They were presented to the Council October 13, 1908, and were referred by the Council to the Committee on Affiliated Societies with the

request to suggest rules to govern such branches. The rules prepared by the Committee were presented to the Council November 10, 1908, and were approved December 4, 1908.

R 27 The name of this Association shall be "—————— Affiliated with The American Society of Mechanical Engineers;" or, "the —————— Student Section of The American Society of Mechanical Engineers."

R 28 The basis of such an affiliation shall be the independence, autonomy and self-control of each affiliated body under its own by-laws; subject to such limitation as may be set by the Council.

R 29 Each affiliated society shall furnish to the Secretary of The American Society of Mechanical Engineers, satisfactory reports of discussions and business transacted at its meetings; also copies of all papers and addresses presented.

R 30 The American Society of Mechanical Engineers will furnish monthly issues of The Journal to all members of affiliated organizations who are not members of The American Society of Mechanical Engineers, upon the payment by each of two dollars per year. The American Society of Mechanical Engineers will also furnish to the secretary of each affiliated body a certain number of extra copies of The Journal for use at its meetings, the number furnished to be agreed upon in each case.

R 31 Upon recommendation of each affiliated society the President of The American Society of Mechanical Engineers, after conference with the Secretary of the Society, shall appoint a member of the latter body to be Honorary Chairman of each affiliated society, for each year. Such Honorary Chairman shall be, *ex-officio,* a member of the Governing Committee of said affiliated society.

R 32 The Presiding Officer chosen by each affiliated society shall be styled Chairman of the —————— Affiliated with The American Society of Mechanical Engineers; or, Chairman of the —————— Student Section of The American Society of Mechanical Engineers.

R 33 The affairs of the affiliated society shall be managed by a Governing Committee of at least three members, besides the Honorary Chairman.

R 34 The names which designate any committees appointed by the affiliated societies shall be different from the names of corresponding committees of The American Society of Mechanical Engineers.

R 35 The affiliated society shall have the privilege of having printing done by The American Society of Mechanical Engineers at cost. Where the affiliated society desires to publish any papers in local journals or elsewhere, it shall first ascertain that The American Society of Mechanical Engineers does not itself desire to publish such paper and the privilege of priority in publication shall always be the right of The American Society of Mechanical Engineers. The affiliated society shall claim no exclusive copyright upon such papers.

ELECTIONS TO MEMBERSHIP

The following were declared elected to membership in the Society upon the ballot December 5, 1908:

MEMBERS

Alden, H. W., Canton, O.
Anderson, Frederick Paul, Lexington, Ky.
Angstrom, C. J., Worcester, Mass.
Ard, Charles Edgar, Agricultural College, Miss.
Armstrong, F. H., Vulcan, Mich.
Ashton, Walter S., St. Louis, Mo.
Atwood, William S., Montreal, Can.
Ayers, N. B., Dayton, O.
Bacon, F. T. H., New York.
Barbieri, Cæsar, Chicago, Ill.
Barnes, William O., Quebec, Canada.
Bartlett, Charles Howard, Medford, Mass.
Bliss, Edwin C., Providence, R. I.
Bruff, Charles E., New York.
Chace, William W., Cleveland, O.
Coleman, William W., Milwaukee, Wis.
Comstock, Charles W., Denver, Col.
Connet, Frederick N., Providence, R. I.
Craig, Charles H., Jr., Needham, Mass.
Dull, R. W., Aurora, Ill.
Dunham, Geo. W., Lansing, Mich.
Dyer, R. A., Jr., Auburn, N. Y.
Eyermann, Peter, Du Bois, Pa.
Farrell, H. C., Beverly, Mass.
Fermier, E. J., College Station, Texas.
Fillingham, M. P., New York.
Foster, W. B., Utica, N. Y.
Freeman, J. Porter, Yonkers, N. Y.
Frost, Edward J., Jackson, Mich.
Gilman, Francis Lyman, New York.
Goodwin, Frank, Ajmere, India.
Gore, Warren W., Beloit, Wis.
Griepe, A. W. H., New York.
Groene, William F., Cincinnati, O.
Hallenbeck, George E., Toledo, O.
Hamilton, C. A., Racine, Wis.
Hansell, William H., Philadelphia, Pa.
Hartwell, Harry, New York.
Harvey, M., Philadelphia, Pa.
Hill, Reuben, Corona, L. I., N. Y.
Hodgins, George S., New York.
Holmes, Joseph A., Washington, D. C.
Howarth, Harry A. S., New Haven, Conn.
Johnson, Louis L., New York.
Kean, A. J. A., Zamora, Mexico.
Keil, Gustave B., Chicago, Ill.
Keller, Joseph F., New York.
Kemble, Parker H., Thompsonville, Conn.
King, George C., Massillon, O.
Latham, H. M., Worcester, Mass.
Lees, John W., Indiana Harbor, Ind.
McBride, Thomas C., Philadelphia, Pa.
McKee, Robert A., Milwaukee, Wis.
McKeen, William R., Jr., Omaha, Neb.
Macfarlane, James, La Boca, Panama.
Manton, Arthur Woodroffe, Long Island City, N. Y.
Maxwell, M. C., Brooklyn, N. Y.
Mead, Daniel W., Madison, Wis.

Mitchell, C. J., Beloit, Wis.
Moore, Edwin A., Camden, N. J.
Muhlfeld, John Erhardt, Baltimore, Md.
Nichols, Charles Hart, New York.
Nordberg, Carl Victor, Butte, Mont.
Paine, Sidney B., Boston, Mass.
Parker, L. S., New York.
Paul, Charles E., State College, Pa.
Perks, George W., Springfield, O.
Porter, John A., Macon, Ga.
Pulman, Thomas C., New York.
Robinson, A. L., Culebra, C. Z., Panama.
Saltzman, Auguste L., New York.
Scott, Arthur Curtis, Austin, Tex.
Seager, James B., Lansing, Mich.
Sibley, Frederick H., Cleveland, O.
Smith, Harry Ford, Lexington, O.
Spiro, Charles, New York.
Symonds, G. P., New York.
Thomas, Carl C., Ithaca, N. Y.
Thrall, George C., Detroit, Mich.
Titcomb, George E., Philadelphia, Pa.
Voight, Henry G., New Britain, Conn.
Waern, A. W., Bethlehem, Pa.
Wallichs, Adolph Otto, Aachen, Germany.
Winship, Walter E., New York.
Wintzer, R., Cudahy, Wis.
Worth, C. C., New York.
Wuerfel, George D., Toledo, O.

PROMOTION TO MEMBER

Alford, Leon P., New York.
Allen, Albert M., Cleveland, O.
Bowen, Harrison S., Chicago, Ill.
Crouch, Calvin H., Grand Forks, N. Dak.
Hall, Rodney D., Buffalo, N. Y.
Knight, G. L., Brooklyn, N. Y.
Libbey, J. H., Boston, Mass.
Ohle, Ernest Linwood, Iowa City, Ia.
O'Neil, F. W., New York.
Perry, Frank B., Allston, Mass.
Richmond, Knight C., Providence, R. I.
Sangster, Andrew, Sherbrooke, Canada.
Wilkins, I. Chester G., New York.
Willcox, George B., Saginaw, Mich.
Van Winkle, Edward, New York.
Woodwell, Julian E., New York.

ASSOCIATES

Bronson, Edward L., Waterbury, Conn.
Bruckner, Rudolph E., New York.
Bump, Archie E., Boston, Mass.
Chester, C. P., Morenci, Ariz.
Damon, John H., Plymouth, Mass.
Eaton, Henry C., Waltham, Mass.
English, H. K., Schenectady, N. Y.
Flanders, Ralph E., New York.
Ketchum, Samuel, New York.
Lindberg, F. A., Chicago, Ill.
Lyon, Jesse D., Altanta, Ga.
Maclaren, J. G., Weehawken, N. J.
Mees, Curtis A., Charlotte, N. C.
Merkt, G. A., Marlboro, Mass.
Moon, Hartley Allen, Birmingham, Ala
Neilson, Frederick C., Hartford, Conn
Pearsall, G. H., Chicago, Ill.
Peck, Henry W., N. Rochester, N. Y.
Ritchie, Francis P., Montreal, Can.
Satterfield, Howard E., Indianapolis, Ind.
Sheperdson, J. W., Johnstown, Pa.
Sloan, Alfred P., Jr., Newark, N. J.
Spencer, F. C., Chicago, Ill.
Taylor, Wyatt W., New York.
Thompson, Byron L., Syracuse, N. Y.
Tomlinson, C. E., Syracuse, N. Y.
Wallace, Jacob H., Boulder, Colo.

PROMOTION TO ASSOCIATE

Cox, F. G., Chicago, Ill.
Douglas, C. C., Boston, Mass.
Earle, S. B., Clemson College, S. C.
Gordon, Rea M., Syracuse, N. Y.
Soper, Ellis Clark, S. Pittsburg, Tenn.
York, Robert, Pine Bluff, Ark.

JUNIORS

Adams, Kilburn E., Boston, Mass.
Alger, Harley C., Saginaw, Mich.
Autenrieth, George C., New York
Bannon, Leo M., Central Falls, R. I.
Baxter, Burke Morgan, Cleveland, O.
Beecher, H. W., San Francisco, Cal.
Beers, Leroy F., Rochester, N. Y.
Bixby, William Peet, Woburn, Mass.
Brown, Howard Hayes, New York
Brown, Richard P., Philadelphia, Pa.
Charavay, Marius A., Jersey City, N. J.
Church, Elihu Cunyngham, New York.
Cole, Cyrus L., Chicago, Ill.
Crawley, George E., New York
Crowell, William J., Jr., Lebanon, Pa.
Day, Irvin W., New York.
De Ved, Horace Warren, Mt. Vernon, N. Y.
Diserens, Paul, Urbana, Ill.
Durfee, Walter C., 2d, Jamaica Plain, Mass.
Emswiler, J. E., Ann Arbor, Mich.
Holl, Charles Ludwig, Milwaukee, Wis.
Hussey, C. W., Yonkers, N. Y.
Hutton, Mancius Smedes, New York
Jealous, Arthur R., Lawrence, Mass.
Keller, W. H., Philadelphia, Pa.
Kennedy, William Arthur, Providence, R. I.
Lapat, Leopold, Paterson, N. J.
Laton, Thomas Jefferson, Durham, N.H.
Leeper, Ralph W., Marysville, O.
Lothrop, Marcus T., Syracuse, N. Y.
McFarlan, Edward, Brooklyn, N. Y.
Monks, William Douglas, Mt. Vernon, N. Y.
Moore, Charles R., W. Lafayette, Ind.
Murray, Arthur F., Harrisburg, Pa.
Neely, F. H., Wilkinsburg, Pa.
Nibecker, Karl, Glen Mills, Pa.
Norden, Carl L., Brooklyn, N. Y.
Pape, J. O., Tipton, Ind.
Rattle, Paul S., Chicago, Ill.
Ripsch, Charles William, Dayton, O.
Robbins, John L., Pittsfield, Mass.
Shodron, John G., Milwaukee, Wis.
Slauson, Harold Whiting, New York
Smith, Mark E., Erie, Pa.
Staude, E. G., Minneapolis, Minn.
Stevenson, T. Kennedy, New York
Stewart, Charles Edward, Tufts College, Mass.
Stillman, Edwin A., New York
Thomas, Albert R., St. Louis, Mo.
Ulbricht, T. Carlile, Brooklyn, N. Y.
Watt, William, Jr., Pietermaritzburg, Natal, South Africa
Whitehurst, Herbert Clinton, New York.
Wicks, H. B. Priestley, Scotia, N. Y.
Wilkinson, Cecil T., Eden, Tex.

SUMMARY

Election to full membership	87
Election to Associate grade	27
Election to Junior grade	54
Promotion to Member grade	16
Promotion to Associate grade	6
Total number declared elected	190

No. 1208

THE CONSERVATION IDEA AS APPLIED TO THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

PRESIDENTIAL ADDRESS 1908

By M. L. Holman, St. Louis, Mo.
President of the Society

The rate at which the natural resources of the United States are being consumed caused the President to call a conference of the Governors of the States in 1908 to consider the questions of the conservation and use of the great fundamental sources of wealth of the Nation. That the question is considered a vital one is shown by the fact that the conference marks the first time in the history of our country when the Governors of the States have assembled at the White House to consult with the Chief Executive of the Nation.

2 Invitations to participate in the Conference were issued to about seventy-one "National organizations concerned in the development and use of" natural resources, and to "the Senators and Representatives in Congress; the Supreme Court; the Cabinet; and the Inland Waterways Commission." With the Governors were three men from each State chosen as advisors.

3 The Conference assembled in the East Room of the White House, Wednesday, May 13, 1908, and the morning session was taken up by the address of the President on Conservation as a National Duty. During the sessions of Wednesday and Thursday the following papers were presented:

- The Conservation of Ores and Related Minerals, by Andrew Carnegie.
- The Waste of Our Fuel Resources, by Dr. White.
- The Natural Wealth of the Land and its Conservation, by James J. Hill.
- Soil Wastage, by Prof. T. C. Chamberlain.
- Forest Conservation, by R. A. Long.
- Resources Related to Irrigation, by George C. Pardee.
- Grazing and Stock Raising, by H. A. Jastro.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

4 A limited general discussion was given on each of these papers. The immediate result of the Conference could only be an expression of opinion, in the form of resolutions. Pursuant to a motion the president appointed a Committee on Resolutions, consisting of the Governors of South Carolina, Utah, Wisconsin, Louisiana and New Jersey.

5 The four National Engineering Societies were represented by their presidents, who formed a part of the audience, and who jointly presented resolutions which went, with others, to the Committee on Resolutions. The object sought by these representatives was a broad treatment of the subject on non-political lines, and an avoidance of any indication of departmental jealousy. During the discussion it became apparent that some effort would be required to keep the Conference from political bias. On Friday morning the Committee on Resolutions being ready to report, the reading of the papers set for Friday was suspended and the resolutions prepared were introduced. The President presided and under his skilful parliamentary guidance the resolutions were adopted without a dissenting vote.

6 The report of the Committee has not been given the publicity which it deserves and I therefore give it in full, as published in the Cincinnati Inquirer of May 16, 1908:

We, the Governors of the States and territories of the United States of America in Conference assembled, do hereby declare the conviction that the great prosperity of our country rests upon the abundant resources of the land chosen by our forefathers for their homes and where they laid the foundation of this great nation.

We look upon these resources as a heritage to be made use of in establishing and promoting the comfort, prosperity and happiness of the American people, but not to be wasted, deteriorated or needlessly destroyed.

We agree that our country's future is involved in this; that the great natural resources supply the material basis upon which our civilization must continue to depend, and upon which the perpetuity of the nation rests.

We agree, in the light of facts brought to our knowledge and from the information received from sources which we cannot doubt, that this material basis is threatened with exhaustion. Even as each succeeding generation from the birth of the nation has performed its part in promoting the progress and development of the republic, so do we in this generation recognize it as a high duty to perform our part, and this duty in large degree is in the adoption of measures for the conservation of the natural wealth of the country.

We declare our firm conviction that this conservation of our natural resources is a subject of transcendant importance, which should engage unremittingly the attention of the nation, the State and the people in earnest coöperation. These natural resources include the land on which we live and which yields our food; the living waters which fertilize the soil, supply power and form great avenues of commerce; the forests which yield the materials for our homes, prevent erosion

of the soil, and conserve the navigation and other uses of our streams; and the minerals, which form the basis of our industrial life, and supply us with heat, light and power.

We agree that the land should be so used that erosion and soil wash should cease; that there should be reclamation of arid and semi-arid regions by means of irrigation, and of swamp and overflowed regions by means of drainage; that the waters should be so conserved and used as to promote navigation, to enable the arid regions to be reclaimed by irrigation and to develop power in the interests of the people; that the forests, which regulate our rivers, support our industries and promote the fertility and productiveness of the soil, should be preserved and perpetuated; that the minerals found so abundantly beneath the surface should be so used as to prolong their utility; that the beauty, healthfulness and habitability of our country should be preserved and increased; that the source of national wealth exists for the benefit of all the people, and that the monopoly thereof should not be tolerated.

We commend the wisdom of the President in sounding the note of warning as to the waste and exhaustion of the natural resources of the country, and signify our appreciation of his action in calling this Conference to consider the same, and to seek the remedies therefor through coöperation of the nation and the states.

We agree that this coöperation should find expression in suitable action by the Congress within the limits of and co-extensive with the national jurisdiction of the subject and complementary thereto by the Legislatures of the several States within the limits co-extensive with their jurisdiction.

We declare in convention that in the use of the national resources our independent states are interdependent and bound together by ties of mutual benefits, responsibilities and duties.

We agree in the wisdom of future conferences between the President, members of Congress and the Governors of the States, regarding the conservation of our natural resources with the view of continued operation and action on the line suggested, and to this end we advise that from time to time, as in his judgment may seem wise, the President call the Governors of the States, members of Congress and others into conference.

We agree that further action is advisable to ascertain the present condition of our national resources and to promote the conservation of the same. And to that end we recommend an appointment by each State of a commission on the conservation of natural resources, to coöperate with each other and with any similar committee on behalf of the Federal Government.

We urge that continuation and extension of forest policies be adopted to secure the husbanding and renewal of our diminishing timber supply, prevention of soil erosion and the protection of head waters, and the maintenance of the purity and navigability of our streams. We recognize that the private ownership of forest land entails responsibilities in the interests of all the people, and we favor the enactment of laws looking to the protection and replacement of privately owned forests.

We recognize in our waters a most valuable asset of the people of the United States, and we recommend the enactment of laws looking to their conservation, to the end that navigable and source streams may be brought under complete control and fully utilized for every purpose. We especially urge on the Federal Congress the immediate adoption of a wise, active and thorough waterway policy

providing for the prompt improvement of our streams and conservation of their water-sheds required for the uses of commerce and the protection of the interests of our people.

We recommend the enactment of laws looking to the prevention of waste in the mining and extraction of coal, oil, gas and other minerals with a view to their wise conservation for the use of the people and to the protection of human life in the mines.

Let us conserve the foundations of our prosperity.

FUNDAMENTAL CONSTANTS

7 As it is the peculiar function of the engineer to assist in making some of the resources of nature available for the use and convenience of man, we may well spend our time this evening considering a few of the phases of the problem. The President particularly desired the coöperation in the movement of the engineers of the United States and subsequently ascribed to the action of the Engineering Societies the credit of inaugurating the conservation campaign on non-political lines.

8 As it was the rapidly increasing rate of consumption of our natural resources that induced the President to make a new departure along the line of "States' Rights" and to call on the Governors for a conference, let us first examine into the forces at work consuming our national inheritance and turn our attention to ourselves, as the primary cause or force that is so rapidly dissipating the natural resources of the nation and emptying the store house of nature. For purposes of the present discussion we will confine ourselves to the territory of the United States. Table 1, compiled from the census returns, gives in a condensed form the fundamental constants and the independent variables of our problem, that is, our lands and inland waters and our population, as they stood in 1900.

TABLE 1 LAND AND WATER, AND POPULATION OF THE STATES OF THE UNITED STATES

State or Territory	Square Miles			Population	Density
	Gross Area	Water	Land		
Alabama	52,250	710	51,540	1,828,697	35.5
Arizona	113,020	100	112,920	122,931	1.1
Arkansas	53,850	805	53,045	1,311,564	24.7
California	158,360	2380	155,980	1,485,053	9.5
Colorado	103,925	280	103,645	539,700	5.2
Connecticut	4,990	145	4,845	980,420	187.5
Delaware	2,050	90	1,960	184,735	94.3

TABLE 1—Continued

State or Territory	Square Miles			Population	Density
	Gross Area	Water	Land		
District of Columbia	70	10	60	278,718	4645.3N.B.
Florida	58,680	4440	54,240	528,542	9.7
Georgia	59,475	495	58,980	2,216,331	37.6
Idaho	84,800	510	84,290	161,772	1.9
Illinois	56,650	650	56,000	4,821,550	86.1
Indiana	36,350	440	35,910	2,516,462	70.1
Indian Territory	31,400	400	31,000	392,060	12.6
Iowa	56,025	550	55,475	2,231,853	40.2
Kansas	82,080	380	81,700	1,470,495	18.0
Kentucky	40,400	400	40,000	2,147,174	53.7
Louisiana	48,720	3300	45,420	1,381,625	30.4
Maine	33,040	3145	29,895	694,466	23.2
Maryland	12,210	2350	9,860	1,188,044	120.5
Massachusetts	8,315	275	9,040	2,805,346	348.9
Michigan	58,915	1485	57,430	2,420,982	42.2
Minnesota	83,365	4160	79,205	1,751,394	22.1
Mississippi	46,810	470	46,340	1,551,270	35.5
Missouri	69,415	680	68,735	3,106,665	45.2
Montana	146,080	770	145,310	243,329	1.7
Nebraska	77,510	670	76,840	1,066,300	13.9
Nevada	110,700	960	109,740	42,335	0.4
New Hampshire	9,305	300	9,005	411,588	45.7
New Jersey	7,815	290	7,525	1,883,669	250.3
New Mexico	122,580	120	122,460	195,310	1.6
New York	49,170	1550	47,620	7,268,894	152.6
North Carolina	52,250	3670	48,580	1,893,810	39.0
North Dakota	70,795	600	70,195	319,146	4.5
Ohio	41,060	300	40,760	4,157,545	102.0
Oklahoma	39,030	200	38,830	398,331	10.3
Oregon	96,030	1470	94,560	413,536	4.4
Pennsylvania	45,215	230	44,985	6,302,115	140.1
Rhode Island	1,250	197	1,053	428,556	407.0
South Carolina	30,570	400	30,170	1,340,316	44.4
South Dakota	77,650	800	76,850	401,570	5.2
Tennessee	42,050	300	41,750	2,020,616	48.4
Texas	265,780	3490	262,290	3,408,710	11.6
Utah	84,970	2780	82,190	276,749	3.4
Vermont	9,565	430	9,135	343,641	37.6
Virginia	42,450	2325	40,125	1,854,184	46.2
Washington	69,180	2300	66,880	518,103	7.7
West Virginia	24,780	135	24,645	958,800	38.9
Wisconsin	56,040	1590	54,450	2,069,042	38.0
Wyoming	97,890	315	97,575	95,531	0.9
Totals and Averages			2,971,038	76,357,575	25.7

TABLE 2 PUBLIC LANDS, JULY 1, 1908

FROM REPORT OF THE GENERAL LAND OFFICE OF THE UNITED STATES

States	Acres	States	Acres
Alabama	129,713	Montana	46,532,440
Alaska	368,021,509	Nebraska	3,074,658
Arizona	42,769,202	Nevada	61,177,050
Arkansas	1,060,185	New Mexico	44,777,905
California	29,872,493	North Dakota	2,322,150
Colorado	23,696,697	Oklahoma	86,339
Florida	414,942	Oregon	16,957,913
Idaho	26,785,002	South Dakota	6,561,295
Kansas	171,446	Utah	36,578,998
Louisiana	116,249	Washington	4,635,001
Michigan	135,551	Wyoming	37,145,302
Minnesota	1,788,705		
Mississippi	42,791	Total	754,895,296
Missouri	27,480		

9 While dealing with statistics of population, it may not be amiss to call attention to the fact that we are not always governed by the will of the majority. Table 3, from the census returns, shows quite the opposite; in fact it is about an even chance that a successful presidential candidate represents the majority of the voting population:

TABLE 3 PRESIDENTIAL VOTE

Year	Total popular vote	WINNING CANDIDATE		
		Popular vote	Per cent of total	Per cent of population
1828	1,156,328	647,231	55.97	....
1832	1,250,799	687,502	54.96	....
1836	1,498,205	761,549	50.83	....
1840	2,410,778	1,275,017	52.89	14.1
1844	2,698,611	1,337,243	49.55	....
1848	2,861,908	1,360,101	47.36	....
1852	3,138,301	1,601,474	50.90	....
1856	4,053,967	1,838,169	45.34	....
1860	4,676,853	1,866,352	39.91	15.2
1864	4,024,792	2,216,067	55.06	....
1868	5,724,684	3,115,071	52.67	....
1872	6,466,165	3,597,070	55.63	....
1876	8,412,733	4,033,950	47.95	....
1880	9,209,588	4,449,053	48.26	18.3
1884	10,044,985	4,911,017	48.48	....
1888	11,372,299	5,440,216	47.83	....
1892	12,059,351	5,556,918	45.73	....
1896	13,923,202	7,104,779	50.49	....
1900	13,967,974	7,217,810	51.66	18.4
1904	13,513,995	7,620,670	57.13	....

10 It will be noted that President Lincoln was elected by 40 per cent of the popular vote and that President Roosevelt received the greatest per cent of the popular vote since 1828. As new parties are developed the chances of government by the minority become greater, and with a sufficient number of political parties in the field, revolutions will be the order of the day.

HEALTH

11 Of our many resources, the most important one is the health of the people. The safeguarding of the general health and the prevention of preventable diseases form a large field for the practical operation of the conservation idea and will require the best efforts of the State and the Nation, the sanitary engineer and the health officer. The experience of the last decade has shown that prompt action by competent specialists, backed by the power of the Government, will conserve to the State, the community and the citizen, an earning power and working capacity that would otherwise be lost or diminished. The preventive work, particularly in zymotic diseases, must fall to the sanitarian and not to the physician, unless we adopt the Chinese practice of paying our doctors to keep us in good health and stop the honorarium when we fall sick.

12 A sharp distinction must be drawn between protecting people from unnecessary sickness and restoring the sick to health. The medical profession is an ancient and honorable one. It has accomplished a vast amount of good and has furnished a multitude of noble examples of devotion to duty. It is not my purpose to criticise the profession, but to draw attention to the fact that we need and must have comprehensive organizations, State, National, and municipal, which have for their field of work the protection of health. The training of the physician, and particularly his experience, are directed along lines that develop the specialist, but as a general rule executive training and experience are lacking. Our municipal records furnish much evidence to the fact that successful physicians, called from private practice, are totally devoid of executive ability. There have been exceptions, but they only emphasize the rule. Some States and cities now have Boards of Health, but for the most part they are composed of physicians and the work handled is the cure and not the prevention of disease. Less than one-half of the States record all deaths and require burial permits, and but few States keep accurate accounts of epidemics and contagious diseases.

13 Our school children are not properly protected from zymotic diseases. Some work in this direction, however, is now being undertaken by New York City. Our business interests suffer large losses from epidemics and generally work to prevent quarantine and other evidences of the true state of affairs from becoming public. Too much emphasis cannot be put on this part of the conservation problem. The public health is our most valuable resource and its protection and conservation is of the utmost importance.

14 One of the first problems in the field of sanitation is the protection of our rivers, lakes and other sources of domestic water supply from pollution. At present most, if not all, of our inland cities and towns discharge raw sewage into the natural water courses and lakes. One notable city for years discharged its sewage into the margin of a lake and took its water supply from the same place. Typhoid fever, resulting from this arrangement of sewage disposal and water supply, was prevalent and widespread. The general Government, through a National Sanitary Commission, should have intervened for the protection of the citizens of other states and the residents of the city. This condition, in a slightly modified degree, still exists, and the necessity for a sanitary overhauling is evident to any unbiased student. Of late a portion of the sewage has been diverted out of the natural water-shed of the lake, but the conditions are far from safe, and a movement is now on foot to divert the remaining sewage ultimately, under the guise of an improved water-way. In this particular instance the sewage should be purified and discharged at the cost of the municipality producing it. An effort to force this treatment of the case resulted in a "Scotch verdict" and the present conditions may be expected to continue until there is a general movement looking to the protection of our water courses from pollution, or until each community develops a distinct bacillus that can be positively identified. In the matter of protecting our rivers and water courses from pollution, we are far behind the times and should begin our conservation era with laws prohibiting all parties, private and public, from burdening the country with filth that should be handled by the parties producing it, and at said parties' own proper cost and expense.

15 In addition to the protection of our sources of water supply the purification of the water furnished to cities must be given attention. It is an open question to what extent the State should exercise its power in this direction, but there is no question as to the results which an impure water supply entails, and the responsibility, morally if not legally, of the municipality which furnishes polluted water. It is my

opinion that the State should require the municipality, which it empowers to levy taxes, to discharge its full duty to its citizens. When a private water company supplies water it is under obligation to furnish a safe drinking water, just as fully as it would be to furnish adequate fire protection.

16 We have not the time this evening to enter into a detailed discussion of any of the problems involved in water purification, but an examination of the results obtained in the city of Washington for the years 1896 to 1897 is ample evidence to convince anyone of what may be accomplished. Numerous other cases might be cited showing good returns in life and health. The evidence is convincing, and the conclusion certain that the residents of our country suffer much bodily harm and unnecessary loss of life from drinking diluted sewage. Life insurance companies make us pay high rates for our negligence and pile up vast sums of money which are often manipulated to increase our burdens further. The reports of the insurance commissioner of the State of Connecticut show for 19 of the large insurance companies total assets amounting to the enormous sum of $2 718 631 737 and liabilities of $2 591 534 168.

FOOD

17 The next phase of the problem relates to the individual, and is not primarily within the field of the engineer, nevertheless the mechanical engineer has been responsible for a share of the trouble, by way of the cold-storage warehouse. No fault is to be found with the work of the engineer, but he has put into the hands of the people a process that is often made use of to foist on the public edibles not fit for human consumption. The chemist and the chemical engineer have discovered and manufactured anti-ferments until it has become necessary for the Government to ascertain by experiment how much chemical preservative the ordinary citizen could consume, and to require the labeling of manufactured foods. However, the labeling of our catsup bottles with the legend that one-tenth of one per cent of benzoate of soda is contained therein does not seem to deter its use. The rate of abuse now prevalent in some sections in storing undrawn poultry, and in exposing chilled meats in transit from storage to retail butcher shops will soon require the Government to regulate the cold-storage warehouse business. It is not apparent which resources the pure food law was intended to conserve. With the paternal form of government obtaining in some of the European

countries, the health of the individual is the paramount resource. With us, however, "civil and religious liberty" seems to include unnecessary exposure to disease. At the Conference in Washington the preventable disease problem was practically overlooked, perhaps from the fact that no trust seems to be operating in that field.

LAND

18 It is a question which is our chief natural resource, ourselves or our country. I have assumed the citizen as the first, and now come to the second, the land which we inhabit. In defence of this classification it is sufficient to mention the fact that the present development of the United States can be traced to small beginnings on our Eastern coast a few years ago, and while not within the province of the engineer to descant on the "rise and fall of nations," it may not be amiss to call attention to the necessity of keeping alive the sterling qualities and sound principles of the pioneers as a safeguard to our future progress as a republic. Our land may well receive our careful attention and its conservation be thoroughly studied. We shall be face to face in the near future with the fact that there is not sufficient land to support us at our present rate of growth and methods of living.

19 We will leave the Malthusian end of this problem to the political scientist and philosopher and direct our attention to the engineering side of the case. The accompanying tables show some of the more important items for the States and Territories. Foreign possessions

TABLE 4 INCREASE OF POPULATION

Year of census	Total population	Immigration	Per cent increase of population	Per cent of immigration	Foreign born	Per cent foreign born
1790	3,929,214	....	.....	....	....	....
1800	5,308,483	....	35.1	....	....	....
1810	7,239,881	....	36.4	....	....	....
1820	9,638,453	....	33.1	....	....	....
1830	12,866,020	....	33.5	....	....	....
1840	17,069,453	....	32.7	....	....	....
1850	23,191,876	....	35.9	....	2,244,602	9.7
1860	31,443,321	2,598,214	35.6	8.3	4,138,697	13.2
1870	38,558,371	2,314,824	22.6	6.0	5,567,229	14.4
1880	50,155,783	2,812,191	30.1	5.6	6,679,943	13.3
1890	62,622,250	5,246,613	24.9	8.4	9,308,104	14.8
1900	75,568,686	3,687,564	20.7	4.8	10,460,085	13.7

TABLE 5 COMPARISON OF URBAN POPULATION AND AVAILABLE FARM LAND

Year of census	Urban population	Per cent of total	Number of farms	Total acres	Average acres per farm	Per cent increase of farms
1790	131,472	3.4	....	....	.....	....
1800	210,873	4.0	....	....	.....	....
1810	356,920	4.9	....	....	.....	....
1820	475,135	4.9	....	....	.....	....
1830	864,509	6.7	....	....	.....	....
1840	1,453,994	8.5	....	....	.....	....
1850	2,897,586	12.5	1,449,073	293,560,614	202.6	....
1860	5,072,256	16.1	2,044,077	407,212,538	199.2	41.1
1870	8,071,875	20.9	2,659,985	407,735,041	153.3	30.1
1880	11,318,875	22.6	4,008,907	536,081,835	133.7	50.7
1890	18,272,503	29.2	4,564,641	623,218,619	136.5	13.9
1900	24,992,199	33.1	5,739,657	841,201,546	146.6	25.7

In the year 1900, on the basis of 4000 and over, the percentage of urban population was 37.3.

are not included, as they do not enter into the problem except as disturbing elements, and it is to be hoped that the franchise will not be extended to them. We must fight out our political differences at home and not wait for returns from the Philippines to know who is to be our next President.

20 Texas became one of the United States in 1845. If we assume the growth in population of the States and territories from the year 1850 to the year 1900 as an indication of the rate of growth which we may expect, that is, judge the future by the past, we can develop

TABLE 6 AGRICULTURAL POPULATION

Year of census	Population engaged in agricultural pursuits	Per cent of total population	Average value of farms per acre	Per cent increase in value
1880	7,713,875	44.3	$22.72	36.2
1890	8,565,926	37.7	25.81	32.0
1900	10,381,765	35.7	24.39	27.6

a formula that fairly represents the case. Representing the population at any time, T years subsequent to 1850, by P, we find that the expression:

$$\operatorname{Log} P = 1.702 (\log T - 0.897)$$

gives the approximate results shown in Table 7.

21 We see that the formula gives results corresponding with the census for the years 1850, 1890, and 1900. The computed results

TABLE 7 COMPARISON OF COMPUTED AND ACTUAL RATES OF GROWTH

Year	Population as shown by census, millions	Population as given by formula, millions	Error of formula, millions excess
1850	23	23	....
1860	31	32	1
1870	39	41	2
1880	50	52	2
1890	63	63	....
1900	75	75	....

for 1860, 1870, and 1880 are high, but the effect of the Civil War may be assumed to be responsible for a large part of the loss. Now *if* this formula holds good, as a rough approximation, for the next hundred years we may expect the growth shown in Table 8.

TABLE 8 ESTIMATED FUTURE POPULATION OF THE UNITED STATES

Year	Estimated population millions	Estimated density per square mile	Year	Estimated population, millions	Estimated density per square mile
1910	89	30	1960	168	57
1920	103	35	1970	186	63
1930	118	40	1980	202	68
1940	134	45	1990	225	76
1950	150	51	2000	245	83

22 The formula is, in my opinion, conservative and gives results that will probably be exceeded up to 1950. Beyond that time the reverse may be true. In round numbers, we may expect a population of 150 000 000 by the year 1950 and at the end of the present century we will be near the 250 000 000 mark. Our development will be both agricultural and industrial, and our limiting density of population will be governed by the ability of succeeding generations to compete in the markets of the world. A superficial study of the data will show that while our density will not become excessive during this century, we will nevertheless have to change our present extravagant pioneering methods of working and living.

23 Free trade would result in rapid changes in density of population, lowering the figures for the manufacturing states and raising them for the agricultural states. This problem is not for the engineer, however, though he will join the hegira while the statesmen are solving the tariff question.

TABLE 9 DENSITY OF POPULATION OF THE NEW ENGLAND STATES

Indicating the possibilities of an industrial development when the market for manufactured goods is protected from foreign competition

States	Total area Square miles	Water area Square miles	Land area Square miles	Population	Density per square mile of land
Maine	33,040	3,145	29,895	694,466	23.2
New Hampshire	9,305	300	9,005	411,588	45.7
Vermont	9,565	430	9,135	343,641	37.6
Massachusetts	8,315	275	9,040	2,805,346	348.9
Rhode Island	1,250	197	1,053	428,556	407.0
Connecticut	4,990	145	4,845	908,420	187.5
Totals and averages			62,973	5,592,017	174.98

TABLE 10 RATE OF GROWTH OF THE NEW ENGLAND STATES

State	1790	1800	1810	1820	1830	1840
Maine	3.2	5.1	7.7	10.0	13.4	16.8
New Hampshire	15.8	20.4	23.8	27.1	29.9	31.6
Vermont	9.4	16.9	22.9	25.8	30.7	32.0
Massachusetts	47.1	52.6	58.7	65.1	75.9	91.8
Connecticut	49.1	51.8	54.1	56.8	61.4	64.0
Rhode Island	63.4	63.7	70.9	76.6	89.6	100.3
Average	31.3	35.1	39.6	43.6	50.2	56.1
	1850	1860	1870	1880	1890	1900
Maine	19.5	21.0	21.0	21.7	22.1	23.2
New Hampshire	35.3	36.2	35.3	38.5	41.8	45.7
Vermont	34.4	34.5	36.2	36.4	36.4	37.6
Massachusetts	123.7	153.1	181.3	221.8	278.5	348.9
Connecticut	76.5	95.0	110.9	128.5	154.0	181.5
Rhode Island	136.0	160.9	200.3	254.9	318.4	407.0
Average	66.6	83.3	97.4	116.4	141.9	175.0

24 The figures are given to emphasize the fact that we have built up a condition in the United States that will require very careful handling to avoid disaster. The tariff has built up the trusts, and the trusts have built up the country, and the change from present conditions must be made slowly to avoid destruction.

25 The density of population of Germany for the year 1900 was 270 per square mile and it has been steadily increasing since that time. A comparison of this figure with the density of population of the

United States will convince the student that the German nation long ago solved most, if not all, of the conservation problems brought to our attention by the President. I venture to suggest that we might make progress by ascertaining the secret of German frugality and prosperity rather than by compiling masses of figures to prove that which is well known, viz: that we are wasting the resources of nature like a true prodigal son.

THE FOREST

26 One of the resources which the pioneer finds at hand and available for immediate use is the forest. It was necessary in many cases to clear away the primeval forests in order to render the land suitable for habitation, and this clearing had to be done at the minimum expenditure of labor. At present, however, the situation is very different. We are consuming over eight times as much lumber per capita as is used in Europe and our timber supply in the older parts of the country has been exhausted. It is high time we were taking thought for the future and making provision for a supply of timber for our successors. A good example of what may be accomplished in this line is to be found in Germany. Our own national Government has been making well directed efforts in this direction, so far as the public lands are concerned, but the States must each work in their respective domains in order to accomplish the best results. The clearing of land for farming is not criticised, but the unnecessary waste of timber resulting from an inordinate greed for gain (stimulated by a tariff) is the thing that needs checking. At this stage of the problem we begin to come into contact with the position of the engineer (I use the word in its broad sense) in relation to this natural resource. He has designed and built the machinery that has made the large production of lumber possible. The slow whip-saw and broadaxe methods of our forefathers have given way to the "shot gun feed," and any ordinary saw-mill will turn out more lumber in a day than the "Pilgrim Fathers" could have produced in a year.

27 The question that the engineer faces in private practice is how to increase the output and decrease the cost per thousand feet board measure. I have personally devised ways and means of *increasing* the fuel consumption of the power plant of a large saw mill, as a means of saving in the operating expenses of the mill. The problem, so far as the mill was concerned, was to produce lumber at the lowest price per thousand feet. A condition, and not a theory, confronted the engineer. The alternative of meeting competition or

going out of business was the problem in this case as in many others, and this same test, under our system of government, is the force that will conserve those of our resources which fall in the class designated by the President as renewable.

28 Some of our manufacturing concerns using timber as a "raw material" are now, and have been for some time past, taking necessary precautions to insure a continuity of business existence. The preservation of forests for other purposes than as a source of supply for timber is "another story," and the questions involved are being worked out by the Bureau of Forestry, under the able direction of Gifford Pinchot. The reservation of forest lands by the Government needs careful handling to insure the protection of the citizen and to see that our timber supply is not captured *en masse* in the future by consolidated capital. A large surplus of capital, labor or materials, generally makes trouble, and care must be taken to see that the national Government does not lay foundations that may be used for a timber trust of gigantic proportions.

29 The waste of timber in the process of manufacture is at present as high as 75 per cent in some cases, and this waste is not counted from the log, but from the dimension lumber. Why is it that an English firm can sell shuttles and spindles in this country, in the face of a tariff? They do not waste anything, and the list of by-products made by them from scrap lumber would stagger a professor. A study of this problem for a New England concern, however, indicated that until the price of logs went up and the price of labor came down, by-products were not profitable undertakings in that locality.

30 The first action reserving public forest lands was due to Act

TABLE 11 FOREST RESERVATIONS

States	Acres	States	Acres
Arizona	13,668,366	Oklahoma	60,800
Arkansas	1,991,899	Oregon	16,221,368
California	25,742,782	South Dakota	1,263,720
Colorado	15,693,157	Utah	7,424,782
Florida	854,891	Washington	12,065,500
Idaho	20,099,029	Wyoming	8,998,723
Kansas	302,387		
Minnesota	294,752	Total	156,707,463
Montana	20,389,696		
Nebraska	556,072	Alaska	12,087,626
Nevada	2,591,052	Porto Rico	65,950
New Mexico	8,474,547		
North Dakota	13,940	Grand total	168,861,039

of Congress of 1891. Since that time the movement has progressed until we now have 168 000 000 acres of national forests in continental United States (See Table 11).

31 We use per capita per year about 500 ft. board measure, as against an average of about 60 ft. for Europe. Our use of timber in house building not only consumes our supply but entails enormous expense for insurance and fire protection. Estimates of the fire tax for the year 1907 run about $2.50 per capita, as against an estimated average of $0.35 for Europe. The total cost for fire losses and for fire protection is not known, but the Government is at work on the problem and may be able to give some estimate in the next census report.

32 Our suggestion for reducing the consumption of timber is by the substitution of fibre products. This is now done to a large extent by the use of paper, and the use can be increased. In Germany peat products are used to replace wood where practical. The waste of fibre-producing material at present is enormous and the sources of future supply practically undiscovered.

WATER

33 Following the President's classification of our natural resources we turn to the one great natural phenomenon which makes our country suitable for the growth of a great nation and the development of a great people. The thing we all must have is water, and for that we are dependent on rainfall. Those of us who have not experienced the hardships of a scant and precarious water supply can have no appreciation of the bounties nature has lavishly scattered through this section of the country, nor of the local conditions created by deficient rainfall in the irrigation regions. In this as in other matters we acquire true knowledge only by hard knocks and bitter experience.

34 The irrigation engineer has to deal with these problems, and to aid in the conservation of the water resources of the country. The people of the irrigation districts, however, are the true promoters of the conservation idea as applied to water, and have made laws which have as their foundation the beneficial use of the water. Necessity has been the teacher and the people have learned their lessons well: as our resources become scarce necessity will teach us the same lesson, and eventually the doctrines of appropriation and application to beneficial use will become general, for the necessities. For the luxuries our present methods will probably remain unchanged.

35 Our present laws and methods of irrigation are of comparatively recent date and are in the formative period, but the beginnings of the art antedate history. Irrigation was practiced by the peoples of ancient China and India and is mentioned in the early Biblical writings. It was evidently practiced by the inhabitants of the arid regions of this country also in prehistoric times. Life could not have been supported without it. Our general practice and methods were first derived from Mexico and show the results of Spanish law and practice. In the United States, irrigation was first practiced by the Mormons. In some parts of the country, irrigation followed mining, but the underlying principles governing the right to use water were fixed by pioneer necessities and followed Mexican practice.

36 The first effort of the irrigation engineer must be to divest himself of his unconscious prejudice in favor of the English law of riparian rights, which is set aside under the conditions which call the irrigation works into existence. The State claims the fee, and grants only an easement known as a water right. Such an easement may be acquired: *first*, by an appropriation, of which some States require a notice to be posted, while a mental process on the part of the appropriator seems sufficient in others; *second*, by the beneficial use of the water. The appropriator acquires the right to only as much water as he can use for beneficial purposes and has not the right to waste water. The process of law required by the State must be complied with and when a decree is issued as a result of the entire process the appropriator becomes possessed of an "adjudicated water right" which is to all intents and purposes property, but which may be abandoned and lost by non-use. Decreed water rights are classified as to priority, and the principle of "first in time is first in right" is strictly enforced. The appropriators of water on a stream are entitled to water in the order of their priorities, which bear the date of the appropriation, and in times of scarcity the last appropriator is the first to be shut down. This shutting down of headgates continues with a growing deficiency of water until in cases of extreme drought the only ditch operating will be the oldest priority. It is the operation of this principle that has led to the storing of water in times of plenty for use during the latter part of the season when water is scarce. The right to take water from the stream for direct application to land during the irrigating season is called a ditch right, the water being conducted from the stream by means of ditches and flumes. Where the water is first conducted into a reservoir either for use during the same season or to be held from one season to another, the right is called a

storage right. Some storage rights are for the non-irrigating or storage season, and others are for the storage of early floods for use later in the same season. In general, water rights are either ditch rights or storage rights. Water rights are classified in accordance with the importance of their use, viz:

Domestic rights, including municipal water supply.
Irrigation rights, ditch and storage.
Power rights.

37 These rights rank as to priority, independent of classification; that is, a power right of the lowest class may have the oldest priority and the first right to take water. A power right, however, can be condemned for a higher use, such as irrigation or domestic use, and in turn an irrigation right may be condemned for municipal use, but the rule does not work in the reverse order.

38 The amount of water taken by an appropriator is regulated and controlled by a head-gate built in the bank of the stream where the water is drawn out, and under the charge of State officials who set them to regulate the rate of flow from stream to ditch. In early days, ditch appropriations were made in inches of water, but the later appropriations, and most of the priorities, are recorded in cubic feet of water per second of time. Some appropriations call for as much water as will flow through a ditch of given dimensions and grade. The inch of water, variously known as miner's inch, sluicing inch and irrigator's inch, is used to designate the rate at which water flows through the head-gate or measuring box. In various states and localities the "inch" represents different rates of flow. While it is an approximate measure, the inch is, nevertheless, a very convenient unit. Many outstanding contracts have the inch as the unit of measurement and decrees are occasionally issued in which the right to water is limited by the number of inches of water decreed. At present the tendency of the engineering profession is to use the "second-foot" and the "acre foot" as the units of rate and quantity. The inch is a handy unit for the irrigator and he will continue to use it, and the engineer must adapt himself to the location and ascertain the rate of flow represented by the local inch. The following memoranda on the inch of water in different states are given for the benefit of members who may need them at some future time:

a The Colorado Statute defines the inch as follows: "Every inch shall be considered equal to an inch-square orifice under a 5 in. pressure and a 5 in. pressure shall be from the top of the orifice of the box put into the bank of the ditch to the surface of the water."

b The Idaho Statute prescribes: "The amount of water that will flow through an orifice one inch square with a 4 in. pressure above the center of the orifice."

c The North Dakota Statute is: "The miner's inch shall be regarded as $\frac{1}{50}$ cu. ft. per second in all cases except when some other equivalent of the cubic foot per second has been stated by contract, or has been established by actual measurement or use."

d In Utah: "Standard measurement of flow, one cubic foot per second, known as the second-foot. Standard measurement of volume the acre-foot."

e The Wyoming Statute is: "A cubic foot per second of time shall be the legal standard for the measurement of water in this State, both for the purpose of determining the flow of water in natural streams and for the purpose of distributing the water thereof."

f Montana: A new act provides that "the cubic foot per second shall be the standard of measurement; 100 miner's inches, equivalent to $2\frac{1}{2}$ cu. ft. per second, not to affect measurements heretofore decreed by court."

g Nebraska: Every inch considered to be one inch square orifice under a 4 in. pressure, and a 4 in. pressure shall be from the top of the orifice in the measuring box to the surface of the water. "Boxes must be 6 in. in height, except where less than 12 in. are delivered; have descending slope $\frac{1}{8}$ in. to the foot and 14 ft. long; 50 miner's inches under 4 in. pressure equivalent to 1 cu. ft. per second of time."

h California: The following is from the San Francisco Post, of the late 70's: "A miner's inch of water is a quantity that will flow through an inch aperture with a free discharge and under a constant pressure of 6 in. above the top of the opening. An aperture of 12.25 in. by 15.75 in., under a pressure of 6 in. above the top of the opening, will discharge 200 in., and is the basis of all measurements where water is retailed in small quantity in the states of California and Nevada."

39 It is customary to express the ratio between the inch and the second-foot as the number of inches which equal a second-foot. The values generally used are:

Colorado, miner's inch.
50 in. = 1 sec-ft; statutory inch: 38.4.
California, miner's inch:
50 in. = 1 sec-ft.
Arizona, miner's inch:
40 in. = 1 sec-ft.
New Mexico, miner's inch:
50 in. = 1 sec-ft.

The inch varies from 50 to the second foot to about 34 to the second foot and the engineer must ascertain the inch in use in the locality for which he is designing work.

40 In some states the duty of water is prescribed by law. By this is meant the duty which a given flow or volume is expected to perform and this is a measure of the quantity which the appropriator is permitted to take for his own use. In states with codes modeled after that of Wyoming, the rate of flow is fixed from 1 sec-ft. for 50 acres to 1 sec-ft. for 70 acres. In Colorado the old standard seems to have been 1 in. per acre, and present practice tends to a flow of 1.44 sec. ft. for 80 acres.

41 Experience has demonstrated the potentiality of the soil of the arid regions, proving that with proper handling good crops can be raised and money made by the farmer. In the Sacramento and San Joaquin valleys there is an irrigated area of 250 000 acres and a population estimated at 300 000 where the population 15 years ago did not exceed 60 000.

42 As an example of what is being accomplished in Colorado, the figures in Tables 12 to 15, compiled from the report of Thos. W. Jaycox, State Engineer, may be of passing interest to us and of value to the irrigation engineer.

43 The most striking feature of the conservation of rainfall is the impounding of the surplus water of one season against the deficiency of another. The work of the Western pioneers has accomplished wonders in making waste places habitable and in building up a territory otherwise barren and unproductive. The Reclamation Service has done good work in opening up for settlement a large amount of land. Table 16, compiled from the Government records, shows the work in hand.

44 The work accomplished by the Reclamation Service is a good example of the conservation idea properly applied, and the example should be followed by States and individuals where practicable. The work of the Government in this line can not be over-estimated; it adds to our resources and that in a way that puts the land in the hands of the farmer.

Irrigation Statistics of Colorado 1906

TABLE 12 EXTENT OF IRRIGATION

	Division No. 1	Division No. 2	Total
Length of Main Ditch, Miles	3314	2840	6154
Length of Laterals, Miles	1927	2147	4074
Water used, Acre feet	1,228,821	1,607,292	2,836,113
Crops Irrigated			
Alfalfa	225,495	136,464	388,959
Natural Grasses	204,907	49,424	254,331
Cereals	328,357	67,051	395,408
Orchards	12,786	14,453	27,239
Market gardens	18,132	2,962	21,094
Potatoes	43,748	799	44.547
Sugar beets	81,375	46,668	128,043
Other crops	34,937	31,913	66,850
Total acres irrigated	921,675	376,734	1,298,409
Total acres that can be irrigated	1,310,125	533,737	1,843,862
Average water applied			
Acre feet per acre			

TABLE 13 YIELD OF IRRIGATED LAND

Boulder County

Fruit	Average yield per acre	Average selling price	Crop value per acre
Apples	300 boxes	$0.90 per box	$270
Cherries	300 crates	2.00 per crate	600
Blackberries	3500 quarts	0.15 per quart	525
Raspberries	4800 quarts	0.12 per quart	596
Strawberries	5000 quarts	0.10 per quart	500

Denver City and County

Fruit	Average yield per acre	Average selling price	Crop value per acre
Apples	300 boxes	$0.90 per box	$270
Plums	500 crates	0.50 per crate	250
Cherries	350 crates	2.00 per crate	700
Blackberries	1800 quarts	0.20 per quart	360
Raspberries	4000 quarts	0.15 per quart	600
Strawberries	2000 quarts	0.10 per quart	200

Jefferson County

Fruit	Average yield per acre	Average selling price	Crop value per acre
Apples	300 boxes	$0.50 per box	$150
Plums	400 crates	0.50 per crate	200
Cherries	350 crates	2.00 per crate	700
Blackberries	1600 quarts	0.20 per quart	320
Raspberries	3200 quarts	0.10 per quart	320
Strawberries	2400 quarts	0.10 per quart	240

Statistics obtained in Denver.

TABLE 14 SUGAR BEET RECORDS

COLORADO, 1907

Acres	Net Tons of beets	Tons per acre	Value of crop	Crop value per acre
16.0	264.70	16.6	$1329.80	$83.11
20.7	393.70	19.0	2000.79	96.65
17.8	541.50	30.4	2633.11	147.93
52.7	1250.00	23.9	6299.57	119.55
30.2	458.60	15.2	2303.44	76.30
9.7	258.40	26.6	1291.79	133.17
48.1	1025.50	21.3	4996.88	103.89
35.5	542.40	15.3	2711.77	76.39
24.0	485.00	20.2	2425.00	101.04
25.0	521.00	20.8	2655.28	106.21
47.0	877.60	18.7	4466.75	95.04
35.0	1028.70	29.4	5143.43	146.95
10.0	301.00	30.1	1504.61	150.46
338.0	5975.60	17.6	29877.90	88.40
47.2	828.40	17.6	4022.85	83.75
17.0	444.10	26.1	2218.99	130.53
37.0	816.70	22.1	4143.73	112.00
50.0	1012.30	20.2	5061.45	101.23
860.9	17025.20	391.10	$85087.14	$1952.60
	Average	21.7		108.46

Average cost of producing is about $35 per acre.

TABLE 15 CROPS WITHIN DRIVING DISTANCE OF DENVER

1907

	Acres	Yield	Selling Price	Value of crop per acre	Net per acre
Strawberries............	14	1600 crates	$1.40 per crate	$588.00	
	25	1600 crates	2.00 per crate	640.00	$464
Raspberries*	12	500 crates	2.25 per crate	562.50	....
	24	1000 crates	1.50 per crate	375.00	....
	35	1600 crates	1.50 per crate	500.00	....
Cabbage, average crop† ..	......	20 tons	0.75 per hundred	300.00	....
Onions..................	......	....	1.50 per sack	300 to 450	180 to 270
Late cauliflower.........	12	....	0.025 per pound	300.00	200
Celery..................	1	3000 dozen	.27 per dozen	810.00	600
Turnips.................	1	20 tons	12.00 per ton	240.00	165 to 190
Parsnips................	1	20 tons	15.00 per ton	300.00	250

* Average cost of picking, crating and marketing $0.55 per crate

† Average cost of production, marketing, etc., $60 per crop

TABLE 16 AREAS, COST, EXPENDITURES, ETC., ON ENTIRE IRRIGATION PROJECTS OR ON SUCH UNITS AS IT IS EXPECTED TO COMPLETE BY 1911

Location	Project	Area in acres	Estimated cost	Estimated expenditure to December 31, 1907	Per cent of completion
Arizona	Salt River	210,000	$6,300,000	$4,362,100	69.2
California	Orland	30,000	1,200,000	16,900	1.4
California, Arizona	Yuma	100,000	4,500,000	1,876,700	41.7
Colorado	Uncompahgre	140,000	5,600,000	2,900,000	51.8
Colorado	Grand Valley	50,000	2,250,000	9,750	0.4
Idaho	Minidoka	160,000	4,000,000	1,839,700	46.0
Idaho	Payette-Bois	100,000	3,000,000	1,381,500	46.5
Kansas	Garden City	8,000	350,000	282,000	80.5
Montana	Huntley	30,000	900,000	796,400	88.4
Montana	Milk River, including St. Mary	30,000	1,200,000	314,800	26.2
Montana	Sun River	16,000	500,000	344,100	69.0
Nebraska, Wyoming	North Platte	110,000	3,850,000	2,797,300	73.0
Nevada	Truckee-Carson	170,000	4,800,000	3,804,600	79.2
New Mexico	Carlsbad	20,000	640,000	579,400	81.5
New Mexico	Hondo	10,000	370,000	358,600	97.0
New Mexico	Leasburg	10,000	200,000	167,900	83.9
New Mexico, Texas	Rio Grande	160,000	8,000,000	53,200	0.6
North Dakota	Pumping, Buford, Trenton, Williston	40,000	1,240,000	519,600	41.9
Montana, North Dakota	Lower Yellowstone	66,000	2,700,000	751,850	64.9
Oregon	Umatilla	18,000	1,100,000	765,500	69.6
Oregon, California	Klamath	120,000	3,600,000	1,305,080	36.2
South Dakota	Belle Fourche	100,000	3,500,000	1,281,900	36.6
Utah	Strawberry Valley	30,000	1,500,000	418,700	27.9
Washington	Okanogan	8,000	500,000	372,180	74.4
Washington	Sunnyside	40,000	1,600,000	481,180	30.7
Washington	Teiton	24,000	1,500,000	565,420	37.6
Washington	Wapato	20,000	600,000	5,220	8.7
Wyoming	Shoshone	100,000	4,500,000	2,313,990	51.5
		1,910,000	$70,000,000	$30,665,570	

From Blanchard

45 Irrigation projects are not proper subjects for the speculator. While there is a large increase in values resulting from a properly executed irrigation undertaking, the increase in value is not an "unearned increment:" the farmer must earn it. The fact that irrigation works are necessary for cultivation does not change the status of the land from that of farm land in general. The real development and improvement must be made by those fitted to till the soil and become permanent inhabitants of the reclaimed land. The inability

to see this simple truth has caused numerous speculators to lose money and has thrown a large number of highly advertised irrigation schemes into bankruptcy. Speculative interests will take desperate chances and trust to the ability of sales agents and glaring advertisements to "unload" before the inevitable failure comes. The reclamation act has avoided this rock, on which so many private undertakings split, by limiting the amount of land to each settler, and the cost per

TABLE 17 CONDITION OF LAND AND EXTENT OF WATER SUPPLY IN THE WESTERN PUBLIC LANDS

States and Territories	Millions of Acres							
	Public land	Grazing	Wood-land	Forest	Desert	Arable	Irrigated	Water Supply
Arizona	72	38	14	5	15	0.2	0.2	2
California	99	20	19	19	20	15.0	1.5	17
Colorado	66	40	9	5		2.0	1.2	8
Idaho	54	20	12	11		1.0	0.5	5
Montana	93	56	7	12		1.0	0.8	11
Nebraska	49	25	2			22.0		2
Nevada	70	42	6	1	20	1.0	0.5	2
New Mexico	78	57	12	4		0.5	0.2	4
N. Dakota	45	38	1			6.0		2
Oregon	60	18	9	19		5.0	0.3	3
S. Dakota	49	38	1			10.0		2
Utah	52	18	16	4	10	2.0	0.5	4
Washington	43	9	13	13		2.0	0.1	3
Wyoming	62	39	3	4	5	1.0	0.5	9

TABLE 18 SWAMP LANDS

States	Area in square miles	States	Area in square miles
Florida	29,000	Indiana	1,250
Louisiana	15,000	New Jersey	900
Western States	10,000	New Hampshire	600
Arkansas	9,000	Massachusetts	500
Mississippi	9,000	Maryland	500
Michigan	7,500	Iowa	400
Minnesota	6,000	Vermont	400
Wisconsin	4,500	Nebraska	400
Maine	4,000	North Dakota	375
North Carolina	3,750	South Dakota	375
Georgia	3,750	Kentucky	350
Illinois	3,500	Pennsylvania	300
Missouri	3,000	Kansas	250
South Carolina	2,750	Connecticut	100
New York	2,500	Delaware	50
Alabama	1,750	Rhode Island	30
Virginia	1,600		
Tennessee	1,250		125,880
Ohio	1,250		

acre of the project, and by requiring the actual use and improvement of the land to acquire title. The act recognizes the fundamental principle of appropriation in the following language:—"That the right to the use of water acquired under the provisions of this act shall be appurtenant to the land irrigated, and beneficial use shall be the basis, the measure and the limit of the right." This provision of the act has been carefully drawn for the protection of the farmer and the confusion of the speculator.

46 The States that have lands subject to their control should follow this good example by adopting laws controlling the development of irrigation projects and exercise stringent control and supervision over water rights and districts subject to floods. In fact, the general government should assume supervisory control over all projects for the development of lands liable to disastrous flooding. Table 17, taken from the Government records, shows in condensed form the general situation and the work remaining to be done.

SWAMP LANDS

47 Another field of operation is the reclamation of swamp lands. A little work is in progress in this line, but compared with the total amount of work to be done, it is not worth mentioning. Table 18, compiled from records of the Geological Survey by Herbert M. Wilson, shows the field for work.

POWER—WATER

48 In connection with irrigation and municipal water supply in arid and semi-arid regions is the closely related problem of power. In all the states in which appropriation of water is made, the appropriation for power purposes is recognized. As a general proposition, the development of a water-shed for irrigation and power purposes, jointly, is not practicable. In sections of the country where coal is cheap, the power phase of the water supply question will lie dormant until the cost of coal reaches a point that makes the installation of hydro-electric plants feasible from a money standpoint. There seems to be a hypnotic power attached to hydraulic power schemes that causes promoters and investors to lose sight of the fact that the real criterion is fixed charges *versus* operating expenses. It may also be the idea that they are going to get something for nothing which warps

judgment. Be that as it may, the fact remains that over-enthusiasm in the direction of hydraulic power plants has caused much trouble and loss of money. In locations such as the Pacific Coast, with coal at about $14 per ton, the hydro-electric plant finds its true habitat, but in the vicinity of coal mines, where cheap fuel prevails, the steam-driven plant is the present solution. So far as the conservation of water power is concerned, much may be said, but the problem, involving as it does the fundamental question of ownership by the individual, the state or the nation, and the further legal question of riparian rights as opposed to appropriation, is one that it will require much study and many legal battles to solve.

49 The water power problem does not form a part of the conservation problem. The question is not one of the waste of water and the resource does not fall into either of the classes into which the President divided our natural resources. It is neither renewable nor non-renewable and its use does not diminish the supply. The question involved is one of ownership and the real conservation will be effected by legislation that will protect the public from the loss of valuable rights. The improvement in long distance transmission from the 10 miles of 10 years ago to the 200-mile lines of to-day makes water power available over large districts. In California four power companies, with an aggregate capital of some fifty millions of dollars control thirty hydro-electric and steam power plants. In the entire state about 250 000 h.p. is in operation under private control.

50 Mr. Lindsay states that there is a potential development of 800 000 h.p. on four rivers in the northern part of California that are under private control and that but 20 000 is utilized, the balance being tied up by speculative interests. Heretofore Congress has required no compensation for water power grants and during the last ten years has handed over to promoters and speculators 33 such grants *gratis*. The same authority states that of about 16 000 000 h.p. now in use in the United States, less than $\frac{1}{4}$ is from water power and 1 600 000 h.p. is going to waste over Government dams. The total water horse-power available for immediate use is given as 25 000 000.

51 I am well aware that the Waterways Commission was "tacked" on to the Conservation movement, but that was because Congress and the President had some differences of opinion regarding appropriations, and the water power question was also included. A strong veto message would stop a water power scheme for which the grantees were not required to render a just equivalent, without the consideration of a National Commission on Conservation of Natural Resources.

52 So far as our Western rivers are concerned, time has shown that the method of transportation which gives the lowest total cost of moving freight is the one that gets the business. For short hauls between river towns water transportation is cheaper and quicker than railways, for ordinary freight, but the advantage disappears as the distance increases. For export shipments, from St. Louis, the additional expense for insurance and interests on cargoes, with the cost of returning empty barges, has proved too great a handicap for river transportation to overcome. These items of expense are not materially lessened by river improvement, and it is not readily apparent how any depth of channel will materially lessen them or reduce fixed charges.

53 As the country becomes more thickly settled our rivers will have to be improved and controlled, and the sooner this work is inaugurated on proper lines the less will be the total cost of the work, but to put forth unsound arguments and to use the public works of the country to influence votes is not conducive to the continued development of our Republic. There is, at present, a strong tendency towards bureaucratic development that is inimical to the successful continuity of our form of Government. The waterways movement is not free from political influences and The American Society of Mechanical Engineers will do well to keep all political questions out of its proceedings.

POWER—COAL

54 Following logically after water-power is the problem of conserving our present great source of light, heat and power, *coal.* To the discovery and use of coal is to be attributed, more than to any other cause, the rapid development of the United States. It is the primary source of the power that has moved the wheels of commerce and manufacture. The rate at which we are consuming this heritage from Nature is sufficient evidence of the necessity of an inquiry into the probable supply.

55 Mr. Wilson, of the U. S. Geological Survey, estimates that our present rate of increase in the consumption of coal indicates that our supply will be exhausted in about 200 years. Estimates in this field, as in other long range predictions, are to be taken, however, as an integration of our present available differential coefficient. The law of the curve may not be properly expressed by deductions from our present knowledge, and future observations may prove that we are

drawing conclusions from insufficient data, obtained at the apex of the curve. I well remember that our professor of mining engineering explained to the class of which I was a member the uselessness of prospecting for coal in the Rocky Mountains. It is quite plain at this time that his data were insufficient. We now realize that the conclusions of a Conference on the Conservation of Whales, had one taken place in New Bedford some years ago, would have created unnecessary alarm and expense. While whales were an important item in that day and generation, the present civilization does not need them.

56 The coal problem is, however, one requiring some consideration. Coal once consumed can not be replaced, and the steadily increasing cost of fuel must be faced. We have in the United States prices ranging from less than $1 to over $14 a ton for coal. I know of a railway power house, in Illinois, where the coal costs about $0.70 a ton delivered on the grates. The only way to conserve our supply of coal is to use it as economically as possible.

57 In the economical use of coal we have two main questions, mining and burning. With our easily mined Western coals it pays to use "wasteful methods" and these mining methods will continue until there is a radical change in present conditions. The mine operator and his engineer must meet competition. They do not adopt mining methods in advance of the times for fear of the certain end of their coal mining operations in bankruptcy. Just so long as it pays to work a mine for the best coal only, this process will continue and the future will have to take care of itself. Which one of you, as householder or engineer, will put up with a poor run of coal, in order that posterity may have good coal? The departments of our Government demand the best grade and are not willing to take the "run of the mine." This means that the poor grades are pushed on smaller purchasers or "go over the dump." If the Government wishes to set an example in the conservation of the coal supply, let an effort be made to adjust the power plants and heating plants in Washington to burn the low grade fuel.

58 Why does not the doctrine of "first in time is first in right," so strenuously introduced and so vigorously supported in the case of "water rights," apply, and permit the present generation to mine as they choose and grant the same privilege to succeeding generations? Who knows but that in the development of the plans of the "Supreme Architect of the Universe" worlds and planetary systems have birth, youth, manhood, old age and death and that we are but a small part

of the ways and means for executing a very small part of the universal plan? Our boasted civilization and rapid progress may be an indication that the end is rapidly approaching.

59 The outlook for improvement in mining, so far as conserving the coal supply is concerned, without a consolidation of competitors is not promising. How to accomplish the desired result under our form of government, with unlimited competition, is not readily apparent; and mining will continue as in the past, until the Statesman and law maker, in deference to the opinions of the people, make radical changes in our fundamental laws. The present waste of both coal and life, in mining, should be reduced to a minimum. Both wastes are due, primarily, to the same cause. The selling price of coal, unless trust-controlled, must be kept down to competition prices, and in the Illinois mines the miner takes desperate risks in order to reduce his percentage of dead work, and occasionally pays for the risk with his life. The unnecessary loss of life has been reduced in other countries and can be reduced in our country. It is due to an uncontrolled greed for gain. Personally, I am of the opinion that the doctrine of "appropriation and beneficial use" offers many advantages not now understood and appreciated, and that it should be, so far as practicable, applied to our remaining natural resources under public control.

60 The mechanical engineer is more interested in the use than in the mining of coal, but here again he is confronted withthe ever-present problem of "making both ends meet" and while he can not afford to be far behind the times, he realizes that he also can not afford to lose money by getting too far ahead of the times. Our last 20 years of progress in electrical lines is a good illustration of the cost of a rapid improvement in the general state of an art, and many of our electrical undertakings are loaded with fixed charges which represent the sins and shortcomings of preceding plants, relegated to the scrap pile before they had earned a fraction of their cost.

61 For the production of power the gas engine and producer give promise of a large saving in coal. The best data at hand indicating the saving that may result in ordinary practice are in the report of tests conducted at the World's Fair of 1903 by the U. S. Geological Department. The condensed results of the tests are shown in the following tables, which represent as nearly as possible the conditions of ordinary practice.

62 At present the gas engine and gas producer are in the stage of development. In the slang of the engineer, there are still some "bugs" to be gotten out of the apparatus. As applied to blast fur-

TABLE 19 COMPARATIVE SUMMARY OF LEADING RESULTS OF COAL TESTS MADE UNDER THE BOILER AND IN THE GAS PRODUCER

Name of sample	Duration of Trial		Total Dry Coal Consumed per Hour*		Dry Coal Burned per Square Foot of Grate Surface per Hour	
	Steam plant Hours	Gas producer plant Hours	Steam plant Pounds	Gas producer plant Pounds	Steam plant Pounds	Gas producer plant† Pounds
Alabama No. 2	10.02	43.00	874	328.7	21.54	7.78
Colorado No. 1	9.97	30.00	722	341.7	17.80	7.56
Illinois No. 3	10.13	30.00	861	356.7	21.23	8.41
Illinois No. 4	10.02	30.00	938	348.5	23.13	7.96
Indiana No. 1	9.93	29.67	908	384.3	22.39	9.08
Indiana No. 2	10.13	7.00	832	312.0	20.51	7.13
Indian Territory No. 1	9.75	31.00	778	374.0	19.17	8.95
Kentucky No. 3	10.07	30.00	882	381.2	21.75	8.92
Missouri No. 2	9.98	4.33	1014	339.6	25.00	7.96
W. Virginia No. 1	9.98	24.00	768	315.6	18.94	7.36
W. Virginia No. 4	10.00	9.00	770	258.2	18.98	5.96
W. Virginia No. 9	10.00	6.33	721	320.1	17.78	7.60
W. Virginia No. 12	10.13	30.00	719	300.5	17.68	6.92
Wyoming No. 2	9.95	30.00	1075	416.5	26.51	9.50

Water Evaporated from and at 212 deg. Fahr. per Pound Dry Coal	B.t.u. per Pound of Dry Coal Used		Electric Horse Power Delivered to Switch-Board		Total Dry Coal per Electrical Horse Power per Hour*	
Steam plant Pounds	Steam plant	Gas producer	Steam plant	Gas Producer	Steam plant Pounds	Gas producer Pounds
8.55	12,555	13,365	213.7	200.6	4.08	1.64
7.21	12,577	12,245	149.1	200.2	4.84	1.71
8.04	12,857	13,041	198.1	199.6	4.34	‡1.79
7.27	12,459	12,834	195.4	198.4	4.80	‡1.76
8.45	13,377	13,037	220.0	199.9	4.13	‡1.93
8.02	12,452	12,953	191.0	201.0	4.35	‡1.55
8.64	12,834	13,455	192.3	204.0	4.04	1.83
8.27	13,036	13,226	208.9	200.5	4.22	‡1.91
7.08	11,500	11,882	205.6	198.6	4.93	‡1.71
8.95	14,198	14,396	196.7	200.4	3.90	1.57
9.65	14,002	14,202	212.5	199.7	3.62	1.29
10.90	14,616	14,580	208.2	201.1	3.46	1.59
9.90	15,170	14,825	203.6	199.8	3.53	‡1.50
5.92	10,897	10,656	182.0	201.2	5.90	2.07

* In gas producer plants this includes the coal consumed in the producer and the coal equivalent of the steam used in operating the producer.

† Coal actually consumed in producer only.

‡ Gas producer hopper leaked during these tests.

nace work and for running on natural gas the gas engine is giving results that indicate its ultimate success. In furnace work the gas engine uses about 40 per cent of the blast furnace gas consumed under boilers supplying steam to a first-class steam-blowing engine of the same capacity. This rate is confirmed by data from a gas engine driven dynamo and indicates that a saving of about 60 per cent in coal is possible by substituting gas engines for steam engines of the ordinary type. On natural gas the guaranteed performance of the gas engine is generally at the rate of one horse power on 10 000 B.t.u. This with "12 cent" gas, in the Joplin lead district of Missouri, will make the fuel cost per kw-hr. on the bus 0.15 cents.

63 With a going concern the operating cost of producing power is generally a small part of the total cost, and the coal cost is still smaller, so that such factors as fixed charges and reliability are very important in deciding the kind of power plant best suited to furnish power for the undertaking. The true solution is found when the cost of power per unit of output is a minimum and there is no diminution of output due to power-house causes. In designing plants, with a high labor and a low material market, the engineer often finds the power plant which will produce a horse-power-hour with the smallest coal consumption to be the most expensive plant under operating conditions. I have in mind from my personal experience two illustrations of this. In one case a power plant and arrangement of machinery were used that would make the graduate mechanical engineer just from school smile in derision, and in the other case the power plant, put in at the request of the owner, who expressed a desire to assist in the improvement of the general state of the art, would delight the heart of a professor of thermo-dynamics. The relative amounts of coal per horse-power-hour in the two plants were about 4 to 1 yet the plant which used 5 lb. of coal per horse-power-hour caused competitors to consolidate for the purpose of reducing expenses while the plant that used $1\frac{1}{4}$ lb. of coal per horse-power-hour fulfilled the desire of the owner, who is still contributing to the advancement of the art and boasting of coal economy. It would not surprise me, however, to get an order at any time to put in a steam plant and dump the gas plant in the scrap pile.

64 In the West our labor market is not yet abreast of the best steam engine practice, to say nothing of the gas engine, and we can only wait until high cost of material shall enable the labor market to furnish the skilled labor necessary to operate the higher grades of power plants.

65 With true British perversity, James Watt adopted 33 000 ft. lb. per minute as the unit of power, and use has fastened this ungainly number on the profession. For this unit we have a good running mate in the German fahrenheit degree. The British thermal unit is the amount of heat required to raise the temperature of 1 lb. of water 1 deg. fahr. This amount of heat is equivalent to about 778 ft. lb. of energy, so that for practical purposes we have 2545 B. t.u. as the equivalent of our unit of work, the horse-power-hour.

66 A pound of coal, ranging from lignites to the best steam coal, will develop an amount of heat ranging from about 9000 thermal units for the lignites to 15 000 thermal units for the best steam coals. If we could utilize all the energy stored in the coal we could produce from 3.54 to 5.89 h.p.hr. per pound of coal.

67 With the steam power plant as a means of transforming the potential heat energy of the coal into power we have, first, the losses due to the boiler plant, which with the most efficient plants will run at least 20 per cent and from that up to 50 per cent and more. Under every-day conditions a loss of 40 per cent may be expected. The next loss is due to the use of steam as a working medium. The theoretical efficiency possible is found by dividing the range of working temperature by the absolute temperature. With the best steam engines a thermo-dynamic efficiency of over 20 per cent is seldom obtained, so that, of the energy of the coal used in a steam power plant we can expect at the best no more than the equivalent of 15 per cent of the energy of the coal. The best efficiency to which I can personally certify is 16 per cent, and that with Illinois coal.

68 In one instance of good practice the heat delivered to the engine was accounted for as follows:

	Per cent
Converted into work	19.6
Radiated from engine	0.76
Rejected to condenser	74.6
Lost in drains from engine	1.71
Returned to boiler	3.33
Total	100.00

69 Good average conditions, however, may be stated for the entire plant about as follows:

	Per cent
Smoke stack gases	22
Radiation from boiler, etc.	7
Radiation from engine and pipe	4
Rejected to condenser	57
Converted into work	10
Total	100

70 Mr. H. G. Stott, Member of The American Society of Mechanical Engineers, estimates the average heat distribution in the power house as shown in Table 20. We may assume that with a good steam generating station we convert but ten per cent of the heat stored in the coal into electricity, on the bus-bars. Further losses due to distribution and conversion, in various ways, to light and power occur so that we get but little of the potential energy provided for our use by a wise and beneficent Nature. Surely, if our ecclesiastical brethren maintain that the storage of coal is a manifestation of Divine Providence, the present inventions for utilizing it must have emanated from his Satanic Majesty.

TABLE 20 AVERAGE HEAT DISTRIBUTION IN THE POWER HOUSE

	Per cent	Per cent
Heat in the coal		100
Loss in ashes	2.4	
Loss in stack	22.7	
Loss from boiler radiation and leakage	8.0	
Returned by feed water heater		3.1
Returned by economizer		6.8
Loss in pipe radiation	0.2	
Delivered to circulator	1.6	
Delivered to boiler feeder	1.4	
Leakage and high pressure drips	1.1	
Heating	0.2	
Loss in engine friction	0.8	
Delivered to small auxiliaries	0.4	
To house auxiliaries	0.2	
Radiation from engine	0.2	
Rejected to condenser	60.1	
Electrical losses	0.3	
Totals	99.6	109.9
Delivered to bus-bar		10.3

71 Another subject which is now engaging a great deal of the attention of advanced engineers is the process of converting coal into

gas in an apparatus called a producer and then burning the gas in the cylinder of a gas engine. This method gives a greater range of working temperature and consequently an outlook for greater efficiency of conversion of coal into work. With the first step of the process, however, the result will average in practice about the same as with the boiler, that is, there is a practicable efficiency of about 70 per cent. In the second step, that is, the conversion of the simple producer gas into power, the outlook for realizing the full benefit of the greater range of working temperature seems to be offset by the cooling necessary to maintain the working parts of the engine. The customary guarantee on gas engine performance is 1 h.p. on 11 000 B.t.u. or about 23.2 per cent efficiency, so that with simple producer gas we have about the same result as with the best steam engine, that is, about 15 per cent of the heat stored in the combustible converted into work.

72 It is quite safe to assume that with our best methods we can not expect more than 15 per cent of the heat value of the coal to be converted into work in the power house. With an efficiency of dis-

TABLE 21 LOSSES IN THE CONVERSION INTO ELECTRICAL ENERGY OF THE POTENTIAL HEAT ENERGY OF ONE POUND OF COAL

SELECTED EXAMPLES SHOWING THE AVERAGE FOR GOOD PRACTICE

With Steam Enriched Gas from Coal of 12 500 B.t.u.	Per cent
Loss in producer and auxiliaries	20.0
Loss in cooling water in jackets	19.0
Loss in exhaust gases	30.0
Loss in engine friction	6.5
Loss in generator	0.5
Total losses	76.0
Converted into electrical energy	24.0

The Westinghouse Company Gas Engine Plant	Plant Per cent	Engine Per cent
Heat in coal	100.00	......
Radiation	10.00	......
Loss in ash	1.00	......
Cooling gases	4.00	......
Boiler fuel	10.00	......
New heat to engine	75.00	100.00
Loss in jackets	26.25	35.00
Exhaust and radiation	27.20	36.30
Engine friction	2.82	3.75
Plant auxiliaries	1.50	2.00
Net work	17.25	23.00

TABLE 21—Continued

Results at the United States Geological Survey Testing Station	Per cent
Losses in producer and auxiliaries	29.0
Losses due to cooling gas, and apparatus	21.6
Losses in exhaust	30.2
Losses due to friction, etc	5.7
Delivered to bus bar	13.5

Analysis of the Buckeye Engine Co.	Per cent
Losses due to producer and auxiliaries	20.0
Losses in cooling water	22.0
Losses in exhaust	32.3
Losses in friction	3.6
Losses in dynamo	1.3
Delivered at switch board, electrical energy	20.8

tribution of 66 to 67 per cent, which, by the way, is good practice, we may get 10 per cent of the total energy of the coal to the customer, but if this is applied with an efficiency of 75 per cent, the customer realizes but 7.5 per cent of the stored energy of the coal. This is good engineering practice at present, and it is safe to say that but few plants are doing as well. We may safely assume the general average as not running over 5 per cent, or in other words we actually use but 5 per cent of the energy of the coal and waste 95 per cent, and, what is more, the outlook for improvement in this direction is not encouraging. In using electricity for lighting we get less than one per cent of the heat value of the coal as useful light. Of all our coal wastes, electric lighting is the greatest.

73 I am fully aware of the possibilities claimed for the gas engine, but my experience thus far, and the practical results with which I am conversant, do not indicate an early fulfillment of the claims. From present indications, the best steam and the best gas engine plants appear to be about on a par with regard to coal economy. The producer plant and the boiler plant are practically equal so far as the first step in converting the coal into power is concerned. The average gas engine is more efficient than the average steam engine, but the best gas engines and the best steam engines are about equally efficient.

74 The city of St. Louis replaced pumping engines that were in first-class running order, with high-duty pumping engines that reduced the coal bills about 75 per cent, but this was not done to conserve

the Illinois coal fields for the next generation. It was good engineering under the load conditions of a large pumping station but in many power plants the installation of the triple or quadruple expansion engine, while saving coal, would increase the cost of the output. I mention this case in order to impress on the rising generation of engineers that interest on first cost must be given due weight in designing plants in order to insure success.

TABLE 22 EXTENT OF THE COAL FIELDS OF THE UNITED STATES ESTIMATED BY THE UNITED STATES GEOLOGICAL SURVEY

States	Square miles	States	Square miles
Montana	47,200	Alabama	8,430
Texas	41,300	Indiana	7,290
Illinois	35,600	Utah	4,580
North Dakota	35,500	Tennessee	4,400
Missouri	23,000	South Dakota	2,400
Iowa	20,000	Virginia	2,120
Kansas	20,000	Arkansas	1,730
Wyoming	19,900	Washington	1,100
West Virginia	17,000	North Carolina	800
Kentucky	16,670	Maryland	510
Indian Territory	14,850	California	280
Pennsylvania	14,680	Oregon	230
New Mexico	13,500	Georgia	170
Ohio	12,660	Idaho	140
Colorado	11,600		
Michigan	11,300	Total square miles	388,940

TABLE 23 COAL OUTPUT FOR THE UNITED STATES FOR 1905

States	Short tons	States	Short tons
Texas	1,200,684	Tennessee	5,963,396
Utah	1,332,372	Kansas	6,423,979
Michigan	1,473,211	Iowa	6,798,609
Montana	1,643,832	Kentucky	8,432,523
New Mexico	1,649,933	Colorado	8,826,429
Arkansas	1,934,673	Alabama	11,866,069
Washington	2,864,926	Indiana	11,895,252
Indian Territory	2,924,427	Ohio	25,552,950
Missouri	3,983,378	West Virginia	37,781,580
Virginia	4,275,271	Illinois	38,434,363
Maryland	5,108,539	Pennsylvania	196,073,487
Wyoming	5,602,021		

TABLE 24 COAL OUTPUT OF VARIOUS COUNTRIES FOR 1905

Countries	Short tons	Millions of tons. Estimated supply, Summersbach, 1904
New Zealand	1,722,379	....
S. A. Republics	2,968,117	....
Spain	3,530,569	....
New South Wales	6,742,186	....
India	9,202,711	....
Canada	8,775,933	....
Japan	11,120,934	....
Russia	21,294,639	40,000
Belgium	24,078,862	20,000
France	37,663,349	19,000
Austria-Hungary	45,209,933	17,000
Germany	191,576,074	415,000
Great Britain	264,464,408	193,000
United States	392,919,341	681,000

TABLE 25 CONSUMPTION OF COAL IN THE UNITED STATES

Year of census	Population	Decade	Coal Consumed	Pounds per capita per year
1820	9,638,453	1816 to 1825	331,356	6.9
1830	12,866,020	1826 to 1835	4,168,149	64.8
1840	17,069,453	1836 to 1845	23,177,637	271.6
1850	23,069,453	1846 to 1855	83,417,825	723.2
1860	31,443,321	1856 to 1865	173,795,014	1105.5
1870	38,558,371	1866 to 1875	419,425,104	2175.6
1880	50,155,783	1876 to 1885	847,760,319	3380.6
1890	62,622,250	1886 to 1895	1,586,098,641	5065.8
1900	75,568,686	1896 to 1905	2,832,599,452	7496.8

75 It will be noticed that the curve of the per capita rate takes on a very rapid rate of increase at 1860. The rate from 1870 to 1900 is approximately represented by the following formula:

$$\operatorname{Log} C = 3.466 (\log T - 0.882)$$

in which T represents the time elapsed in years from 1800, and C represents the pounds of coal consumed per capita per year. The several values resulting from this formula are shown in Table 26.

76 Combining this formula with that previously given for the rate of increase of population, we obtain the following:

$$\left.\begin{array}{r}\text{Log of annual consumption}\\ \text{coal in millions of tons}\end{array}\right\} = 5.168 \log T - 7.673$$

77 Now *if* the rate at present indicated continues, we may estimate our annual consumption of coal roughly, as in Table 27. This total consumption of 614 900 millions of tons of coal would about exhaust the total supply of the country, at the present rate, in the next 100 years. It is not probable, however, that this rate will continue, but that the curve will reach a maximum and then slowly recede, becom-

TABLE 26 CONSUMPTION OF COAL PER CAPITA IN THE UNITED STATES WITH ESTIMATE TO 1950

Year	Per capita consumption of coal per year Pounds	Per capita rate from census reports Pounds
1870	2176	2176
1880	3457	3381
1890	5194	5066
1900	7499	7497
1910	10,400	....
1920	14,090	....
1930	18,630	....
1940	24,040	....
1950	30,550	....

TABLE 27 ESTIMATED ANNUAL CONSUMPTION OF COAL IN THE UNITED STATES

Year	Rate millions of tons per year	Total millions of tons consumed during the decade
1900	460	4,600
1910	750	7,500
1920	1,190	11,900
1930	1,790	17,900
1940	2,620	26,200
1950	3,430	34,300
1960	5,220	52,200
1970	7,460	74,600
1980	9,240	92,400
1990	12,750	127,500
2000	16,580	165,800
	Total	614,900

ing asymptotic to the time axis, when the coal is all consumed or a substitute is found, or something else happens. Prognostications based on the assumption that the population and the coal consumption curves will continue along the paths of the past 40 years must be introduced with a big If, and the if must be given due weight. The study of mathematics and the science of numbers develops a keen

sense of humor, unappreciated by the laity. The law of past increase may hold good for a short time, but it is quite certain that it cannot continue indefinitely. New conditions will obtain and new methods and materials will be introduced; alcohol manufactured from waste vegetable matter is one possible alternative. When our successors can no longer afford electric light and traction they will have to get along without it.

MINERAL RESOURCES

78 Of our mineral resources, iron stands at the head as the distinct feature in our past development along mechanical lines, notwithstanding that electricity is trying to relegate us to the "copper age." An unlimited supply of metal is not needed. We require simply sufficient metal for a working capital and to keep the stock good by supplying for wear and tear. At the present time an immense amount of iron is lying in the wrecks and scrap iron piles all over the country because it does not pay to work it over.

79 The latest estimates, based on visible supply and rate of consumption, indicate an exhaustion of the supply in about 200 years. We waste iron and steel because it is cheap, but as time goes on the waste will diminish and the use be curtailed as the law of supply and demand shall dictate. The most reliable estimate that I have been able to find, places the world's production of pig iron for 1907 at about 61 000 000 tons, of which the United States furnishes about 26 000 000. In 1872 the output of the United States was about 2 000 000 tons. These amounts give us a per capita rate of about 0.05 tons in 1872 and 0.30 tons in 1907. As to the gradually increasing price of iron, we quote Mr. Frank A. Munsey to the effect that the Steel Corporation contracted with the Great Northern Railroad (James J. Hill) to take ore from the Great Northern, "the price to be advanced each year over the preceding year, 3.4 cents. The price for 1897 was $0.85 per ton."

80 The conservation of iron must be by reducing waste to a minimum. As the supply of iron decreases the price will increase and the scrap iron merchant will become an important factor. The mathematical limit appears to point to the control of the old iron market as the ultimate function of the Steel Trust of the future.

81 Of the conservation of oil and natural gas we have neither time nor inclination to discuss the profligate waste in this country, particularly in the South-West where wells have been bored only to

waste the raw petroleum products and the money expended in drilling. All are familiar with the natural gas history of Pittsburg, but the waste in that case is not to be compared with that of the great South-West. The operation of natural laws in the case of these resources has rendered conservation unnecessary in many localities. The only thing that can stop this brand of insanity is a strict application to this class of natural resources of the doctrine of appropriation and beneficial use. The entire community has an interest in natural resources and should not permit waste. This doctrine is not the idle dream of the socialist but is the result of years of experience and is probably the oldest fundamental law on the statute books of the world.

Conclusion

82 Individually the members of the Society will find it necessary in the practice of the profession to design, build and operate works so as to compete with others in the same line. The crucial test will always be found in the cost per unit of output, and the real work of the engineer, in the field of conservation, will be measured by the ability of the works designed by him to compete in the markets of the world. As times and conditions change the engineer must modify his plans and processes or meet with failure. The United States at present is a country of cheap materials and high-priced labor and the result is that it pays to waste material to save labor. In countries where the reverse is found, that is, cheap labor and high-priced materials, it pays to waste labor to save materials. This fact is brought home only to the engineer who has had the opportunity to design work under foreign conditions.

83 With our present high cost of labor we must take all possible precautions to keep the price of materials low. With high-priced materials and high-priced labor we will be shut out of the world's markets. In fact, as it now stands, other countries are under-selling our mills in our own country, by virtue of low-priced skilled labor. The true exponent of the conservation movement should be the skilled labor of the United States.

84 The engineer is also interested, as one of the producing class, in the conservation of the materials of the country and likewise in the conservation of the results of his own labor. The engineering societies must fall in with the conservation idea and see to it that the returns from the societies are commensurate with the effort expended in operating them. Our societies must change with the times and

adopt new methods as the occasion requires. The American Society of Mechanical Engineers has been alive to this fact, as the Journals of the year will show. The credit of the work belongs to the Secretary of the Society, Mr. Calvin W. Rice, and his efficient corps of assistants.

85 In closing I take the liberty of suggesting that the Society take one more step along the road leading to the conservation of the personal funds of its members and at the same time extending its own influence and adding to its own exchequer. This step is the production of an engineer's notebook that will give the practicing mechanical engineer the fundamental constants needed for daily use and enable him to throw several yards of so-called note books from his library. Such a book, published by the Society for the use of its members, would be an incentive to engineers to join the Society and save the young members of the profession enough money to pay for life memberships.

86 I venture to suggest that most of the older members have paid out more money for note books than for dues to the Society and I most earnestly recommend that The American Society of Mechanical Engineers make an effort to conserve both the time and the funds of future members by issuing a note book which shall gradually become the standard mechanical note book, which will be a fund of information for new members, and which will make the Society known wherever engineering is practiced.

No. 1209

THE ENGINEER AND THE PEOPLE

A PLAN FOR A LARGER MEASURE OF COÖPERATION BETWEEN THE SOCIETY AND THE GENERAL PUBLIC

By Morris Llewellyn Cooke, Philadelphia, Pa.
Junior Member of the Society

In the work of the engineer there are three parties interested, i.e., the engineer, his employer and the public. While always recognizing the claim of this third party, engineers as a class have done little, directly, to satisfy it. The chief service rendered the public by the engineering profession has been one rendered indirectly by serving well the second party interested—the employer. It would appear that the time is near at hand when in matters in which they are specially qualified engineers must, individually and collectively, labor for the public interest with just as much fidelity and zeal as they work for their employer, and this not as public spirited citizens but rather as members of a public spirited profession.

2 If the standing of the engineering profession is to grow in the public esteem, it will be because of the effort of the profession to familiarize the public with its aims and ideals. If our own Society, representing the mechanical side of engineering practice, is to develop fully the possibilities of its field, it will be by utilizing every method of making its work of direct value to the people. Primarily this paper seeks to give some of the present day conditions which seem to demand a broadening of the lines of professional activity. A second object is to make a specific recommendation which, if adopted, it is believed will lead along a safe path to a stronger position for the mechanical engineer, considered as a member of the community. In this new position the mechanical engineer will have placed at his command powerful forces at present denied to him and yet absolutely necessary for the full accomplishment of his professional mission.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

THE ENGINEER AS A CONSERVATOR

3 There will probably be little protest against the conception of the engineer as the special conservator of the public interest in matters involving engineering. This view is certainly held by a majority of the leaders of the profession. The roll of our Society has always carried the names of scores of useful citizens who demonstrate in their lives their devotion to the public welfare. In the history of the Society many instances may be found where it has departed from the professional groove to coöperate in some broad way with a section of the general public. Our Society, through the activity of its distinguished President and Secretary, only interpreted the best thought of the membership in its recent valuable service to the National Government in connection with the work of the Conservation Congress and later with that of the Conservation Commission.

4 Granting this much, no one who gives the matter thought will be likely to contest the assertion that in the manifold activities of the Society and of the profession, the general public as such has been almost ignored. Among the announced committees of the Society not one has a title indicating a function broader than the conservation of the interests of engineering and of the engineering profession. Among the thousand and more papers listed in the last published index to Transactions, it is difficult to find a single title which indicates that the author was addressing any one beyond his professional audience. Inquiry among the lay public will show that while our Society is given full credit for its professional strength, absolutely no credit is given to it for interest in public matters. This is the public's estimate of technical organizations as a class.

THE OBJECT OF THE SOCIETY

5 The distinguished authors of the constitution of the Society stated its object in a way to permit of an almost indefinite expansion of function: "The object of the Society is to promote the arts and sciences connected with engineering and mechanical construction." That the constitution, however, imposes no duties on the membership except those of a purely professional interest, witness as follows: "The principal means for this purpose shall be the holding of meetings for the reading and discussion of professional papers and for social intercourse; the publication and distribution of its papers and discussions; and the maintenance of an engineering library." The library may

be made in the future to minister to the requirements of the layman interested in technical matters; but up to this time its direct influence on the great interests of the people has been almost negligible.

6 It can be granted for the sake of progress that the ideals of the engineering profession in this matter are sound. If it has been demonstrated that our practice does not square with these ideals, there appears to be good reason for taking action of some kind. It occurs to the writer that a radical, and in every way salutary, change can be brought about in the Society's relations with the general public by the appointment of a committee to be known as the "Committee on Relations with the Public." During the period required for amending the constitution to provide for such a Standing Committee, its work might be begun by a special committee appointed by action of the Council.

7 The work of a Committee on Relations with the Public would of course have to be done within carefully thought out and definitely prescribed limits. Even so, the work could be made very broad. The Committee would doubtless seek to establish such relations with the lay press as would make its advice and help sought when engineering matters are up for public discussion. It would also doubtless seek to give publicity to the fact that our Society stands ready to offer disinterested advice through its Council to government officials—municipal, state and national. In most cases, of course, this assistance would be not so much to answer engineering questions as to counsel with government officials as to the proper procedure to obtain answers. Such a relation now exists, for instance, between the federal Congress and the National Academy of Sciences. Under the Academy's charter the Congress has a right to call on it for help; and such assistance is at times requested. Such coöperation between governmental agencies and organizations belonging to the architectural profession is not at all uncommon, especially in foreign countries.

POPULAR LECTURES ON ENGINEERING

8 A Committee on Relations with the Public might also provide for courses of lectures for the general public, to be given under the direct supervision of the committee in the Engineering Societies Building in New York, or under the auspices of local engineering societies at other places. This committee might arrange with the Meetings Committee for at least one lecture on an engineering topic of general public interest at each Spring and Winter meeting, which the gen-

eral public would be not only invited but urged to attend. Another important part of the work of such a committee would be to arrange that a fair proportion of the professional papers presented at the various meetings of the Society should be of direct use and value to the public as well as to the mechanical engineer. The cumulative effect of these and of such other lines of work as would soon suggest themselves would be very marked. It is believed that through the work instituted by such a committee the mechanical engineer and the general public could be brought in a few years to a much better understanding.

9 Some members of the Society—especially among those engaged in the more abstruse branches of engineering—never having given much thought to this matter will doubt whether any part of the work of the mechanical engineer can be made of direct interest to the public. Of course, should the Society make coöperation with the public a definite part of its work, it is reasonable to suppose that lines of activity would be opened up which cannot now be foreseen. There is enough work in sight, however, to engage our attention for some time to come.

10 To illustrate: At the annual meeting in December 1905, our distinguished Past-President and present Honorary Councillor, John R. Freeman, delivered his presidential address "On the Safeguarding of Life in Theatres—A Study from the Standpoint of an Engineer." While the work, of which this address was a report, grew out of the Iroquois Theatre fire, it included a comprehensive study of all the great theatre fires during the past 100 years. The work was undertaken at the instance of a prominent manufacturer, two of whose nieces were among the nearly 600 victims of the Iroquois Theatre fire. This public spirited man wanted the investigation made "not for the purpose of fixing the blame but to help us find out how such fearful disasters can be prevented." Mr. Freeman's investigations covered every phase of the question, and in his report he made certain specific recommendations as to theatre construction and the prevention of fires in buildings of this class, such as would appear to reduce the liability to loss of life to a minimum. The recommendations were so logical and relatively so simple that if they could have been made of general public knowledge, theatre owners and proprietors must almost inevitably have been forced by public opinion to their observance even if not impelled thereto of their own accord.

NECESSITY FOR MOLDING PUBLIC OPINION

11 Because this address had a professional interest for but a very small number of mechanical engineers it naturally received comparatively little notice in the technical press. Practically no notice was taken of it in the strictly lay press. Without a certain amount of publicity given to the results, an investigation of this kind might as well not have been undertaken. Mr. Freeman therefore had printed at private expense a few hundred copies of the paper and sent them to privately obtained lists of architects, building inspectors, fire marshals, theatre owners and proprietors, actors and actresses, editors of newspapers, mayors of cities, etc.

12 In the writer's opinion this kind of work should be done by the Society and can be better done by it. The Society should be so organized that matters of public interest may not only be investigated and reported, but that the reports may be given the requisite amount of publicity of the proper kind. Mr. Freeman's paper was filled with matter that almost any magazine or newspaper would have been glad to publish had it been properly shaped up for them when it was still "news." Abstracts of varying length might have been prepared by the editorial staff and published without cost in the lay press. This would have relieved the author of much work entirely outside his line, and not only have secured for the report a fuller measure of usefulness, but would have demonstrated to the public the breadth of our interests and work.

13 From its inception our Society has taken an advanced position in this matter of publicity. We are prohibited by our constitution from copyrighting papers read at our meetings, in order to facilitate their being reprinted in other publications. The constitution further says on this subject: "The policy shall be to give the professional and scientific papers read before it the widest circulation possible with the view of making the work of the Society known, encouraging engineering progress and extending the professional reputation of its members." The Society may some day change the rule which prevents the republication even of abstracts of papers before they are actually presented at meetings. It would seem policy for us to interest the public in as much of our work as can be made intelligible to the public.

THE MOMENTOUS CONSERVATION PROBLEM

14 The matter of the conservation of our national resources will afford probably for years a practically limitless field for investigation

and earnest discussion. The questions involved are so momentous as to warrant the engineer in seeking ways of coöperating with the National and State governments for their solution. No better way could be found for our Society to assist in this work than to provide in its meetings and publications a forum for the discussion of those phases of the general problem in which the mechanical engineer is specially qualified to speak. Many of these questions must be made clear to the layman because legislation will be required for their proper solution.

15 This further step in the line of coöperative engineering effort may seem to some of our members to be a radical one. But it appears to be only the logical sequence of those which have gone before. The records of engineering in the early days of the last century show that engineers had not then learned how to coöperate with each other. Up to a comparatively recent date the professional knowledge of an engineer was his own property and he felt that in imparting any of this knowledge to another engineer he was doing himself an injury. Discussions between engineers were carried on mainly with the object of doing each other harm rather than good. The large number of engineering societies, with the fundamental object of the free exchange of engineering data between their members, show how thoroughly engineers have learned that the general policy of giving freely of their knowledge to their fellow practitioners increases rather than diminishes their effectiveness and, in proportion, their earning power. In proposing the appointment of this new standing committee, the writer suggests only the extension of this coöperation to include the public. There is good reason to believe that engineers have as much to gain from coöperation with the public as they have undoubtedly gained from coöperation with each other.

A SECTION ON PUBLIC MATTERS

16 If the Society once expresses a desire to receive papers of interest alike to the engineer and the layman, the time will probably soon arrive when the increasing number of such papers will suggest the necessity for a Section on Public Matters. Such a section would be conducted like any other section of the Society and would promote and receive papers on subjects of special interest to the layman. These papers and the discussion on them would naturally have to be handled in a way to make them not only intelligible but of practical value to the layman. This work would naturally give rise from time

to time to broad practical summaries covering the essentials of good practice in fields where the general public is directly interested. These publications might easily become of the greatest value to legislators, national, state and municipal, on matters now generally settled without much professional advice or technical knowledge. Such a section would draw to its membership those members of the Society public spirited enough to be willing to devote some time to the investigation and discussion of technical matters in their application to the public interest. It would also probably attract as affiliates many of that rapidly increasing body of laymen who are giving freely of their time and money for the development of the best in the life of the whole people. The work of this section should be given a world-wide scope and interest.

THE AWAKENING IN THE MEDICAL PROFESSION

17 In considering the advisability of adopting the recommendation it is possible and beneficial to study the evolution of the same idea in the medical profession. Not so many years ago it would have been considered in many localities the height of unprofessional conduct for a physician to address any public gathering on a question relating to the public health. This viewpoint has gradually changed until it is almost impossible to pick up a newspaper without finding some evidence of the world-wide propaganda which the medical profession is carrying on for the education of the general public. The movement toward this change in viewpoint was materially assisted and quickened by the tuberculosis campaign. As soon as the medical profession discovered that this disease could probably be exterminated by simple scientific precautions it realized that its obvious duty was to explain this to the public. Under the old standards of professional etiquette this was impossible. So the old standards were changed.

18 In this movement it would be easy to underestimate the influence of the general public. It will be remembered that many of the first steps taken in the fight against the "white plague" were taken by laymen and by organizations made up largely of laymen. It was in a measure this activity of the laymen which forced the medical profession—the natural leaders in such a movement—to assume the lead.

19 Under the new code of ethics it is just as easy for a medical practitioner to discuss publicly the medical inspection of school chil-

dren, or the care of the insane, as to promote the crusade against consumption. For three or four years past the American Medical Association has had in the field a physician who travels over the country for the double purpose of talking to physicians on matters relating to the improvement of conditions in the medical profession and of lecturing to the public on questions of vital importance from a sanitary point of view. These lectures are well attended and are accomplishing much good. Last year the association created a Board of Public Instruction on Medical Subjects. While this Board has done much work it has really only formulated a preliminary scheme of operation. State and county organizations are encouraged to provide lectures, to enlist the interest of the newspaper press, and to watch and influence legislation, wherever it contributes to the welfare of the public.

THE MEDICAL AND CHIRURGICAL FACULTY, FOUNDED 1799

19 The result of this invitation to the local societies can be studied in the activities of almost any medical association. For instance, the Medical and Chirurgical Faculty, the representative medical association of the State of Maryland, an organization which dates its existence from 1799 and one of the most conservative bodies of scientific men to be found in the country, has now many committees solely engaged on matters of public concern. It maintains public lecture courses. It provides the newspaper press on request with signed and unsigned articles on any subject related to the public health. It has recently organized a state-wide campaign for bringing to the attention of the people and the legislature the necessity for a more liberal policy in the care of the insane.

20 As may be imagined, the change in professional attitude reflected by this widespread activity on behalf of the public was not brought about in a day or without powerful opposition. Indeed the writer is informed that some high minded practitioners of advanced years still feel that a mistaken policy is being followed. But the work of practically every medical association—national, state, county and city—indicates that the old conception of the duty of the profession to the public has been permanently abandoned. The president of the American Medical Association, Herbert L. Burrell, M.D., in his annual address of this year, devoted the major part to this subject, saying in conclusion: "A great duty rests on the practitioner of medicine today. He must not shirk it; he must rise to his new burden,

accept it and bear it. The reward to the medical profession for taking this new burden of judicious publicity in medicine will be a broader life for the practitioner, a greater consideration for his fellow man, better citizenship and the recognition by the world that the medical profession is a great public benefactor."

ASSISTANCE NEEDED IN EXPERIMENTAL WORK

21 As the range of human knowledge widens, the expense involved in carrying on the investigation and experimental work necessary to progress in such a profession as ours increases very rapidly. For the engineering profession to keep pace, for instance, with the progress being made today in medical science, our investigators and experimenters must have larger financial resources at their disposal than they have enjoyed in the past. Such foundations and bequests must come for the most part from those unconnected with the engineering profession and from those who put a value on the engineer's work for mankind. Such assistance will result only from a widespread appreciation of the engineer's achievements for the public good and of the requirements of engineering.

22 It is no longer possible for either a profession or a craft to corner information and hold it for its own use. Broadly speaking, those who seek information in any field can obtain it, or at least enough to answer their immediate purposes. And therein lies a danger. This danger would in itself constitute a sufficient reason for the engineer to take the public into his confidence. For after all it is public opinion and not the dictum of the engineering fraternity which finally decides the large questions of engineering practice. How much better it would be then to join forces with the people, to work out with the people the people's problems and to build up in the lay mind such a confidence in our devotion to the people's cause that they will be willing to let us lead in matters where our training especially qualifies us to do so. Only by educating the public to understand and appreciate the work of the engineer, can the public be made to demand the best that can be devised and executed by trained and skillful men. This will have a two-fold beneficial effect on the profession in that it will make more work for the engineer and will give the public that general acquaintance with engineering matters which will make it suspicious of short-cuts.

23 Dr. Hadley, in his address on "The Professional Ideals of the Twentieth Century," delivered at the opening of the Engineering Societies Building, spoke to the engineering profession as follows:

> Yours is the proud boast of having in one brief century established science as the arbiter of the material affairs of mankind, and of having enforced her worship upon a world once reluctant but now gloriously admiring.
>
> Well then, you will ask: Is there anything which remains to be done comparable in importance to this? Yes, there is. An equally large part; perhaps in one sense, a much larger part of your professional duty yet remains to be accomplished. It is not enough to have technical training. It is not enough to know the special sciences on which the practice of a profession is based. A man ought to have a clear conception of the public service on which his profession is based; a man ought to have clear conception of the public service which his profession can render and the public duty which its members owe. Thus, and thus only, can the engineer, the lawyer, the physician or a member of any other learned profession rise to the full dignity of his calling.

BROADER TENDENCIES OF THE DAY

24 There may be some who feel that in launching a movement of the kind proposed the Society would run the risk of getting into difficult positions and even of antagonizing friendly interests. Judging from what is going on around us in this Republic today, and in fact throughout the world, a step of this kind will commend itself so powerfully to the public as to bring to our standard a hundred friends for every one we could possibly lose. However, were this not so there is no sacrifice which the profession ought not to be glad to make to put itself in line with the most advanced ideas of public service. For after all the greatest value to come out of the broader activities is the inspiration to the individual engineer in his everyday work from this closer association with that great employer, the people, for whom in the last analysis all ultimately useful work is done.

DISCUSSION[1]

Dr. Alex. C. Humphreys expressed himself as fully in accord with Mr. Cooke's proposition and arguments, and especially with his contention that this Society should in a definite way recognize and meet its responsibilities to the public.

2 He said in part: As engineers we must bear a definite responsibility by reason of our special training, not only as members of the profession, but as citizens qualified along certain specific lines. As

[1]The discussions on this paper are here given in abstract. They were published in full in the March, 1909, number of The Journal.

individuals this responsibility will vary in direction and measure according to our capacity to advise along certain well-defined lines. . . . . . The responsibility rests upon the engineer, not only to use his knowledge where it can be employed authoritatively for the public good, but to withhold his advice unless his technical training has been adequately supplemented by practical experience. Here come in the advantages accruing if this responsibility to the public is recognized by such an organization as ours, rather than if its recognition is confined to individuals. Advice offered by our Society or by any other engineering society of established reputation would be that which had stood the test of discussion and criticism by the membership at large, or had been carefully canvassed by a competent committee, as suggested by Mr. Cooke.

3 The American Gas Institute has two standing Committees which have general charge of all investigations and the procuring of papers for its conventions. These are the Technical Committees and the Public Relations Committee. Problems of public relations are definitely brought to the attention of the members.

4 The engineers of this country individually, and still more so collectively, are responsible as units of a self-governing people, to guide public opinion aright, wherever they are specifically qualified, and so to guide legislative and executive action.

5 Perhaps we may be more inclined to recognize our responsibility if we constantly bear in mind that as engineers we are not scientists, but that we are charged with the duty of practically applying the truths of science to the solution of the world's industrial problems.

6 Responsibility to the public should be kept constantly before the students of our schools of engineering. Especially should this be a feature of the instruction in the departments of engineering practice; though opportunities for impressing upon students their duty as citizens are not confined to any one department.

Dr. Talcott Williams[1] said that the suggestions in the author's paper are the natural and fortunate product of the joint experience of the mechanical engineer and the journalist. He further said: In it he has pointed out why the calling which has a more severe technical training than any other and gives this age, in the various work and achievement of engineering, its crowning difference from other ages, has less weight in public affairs and on public opinion than any other.

[1] Talcott Williams, LL.D., Philadelphia.

Engineering creates modern life. Take away engineering, and we are what our predecessors were. Add engineering, and the modern world is.

2 Yet modern life pays little attention to the word of the engineer. What engineer here has not seen in some public work or design money wasted today, or waste made sure in the near future, for lack of engineering advice known to engineers? When I became a reporter, nothing amazed me more, after I had read in the technical journals of the brilliant skill with which some engineer had made the impossible, possible, than to go to report the "opening" and find everybody on hand, and sometimes everybody's name on the work, save the one man who had made it famous, but had not made himself known. In every social system, a man or men, a class or a calling will have weight, influence and authority in proportion to its access to the center and origin of authority and power. In a broad democracy, the men who have access to the mass are certain to have weight.

3 The engineer has the disadvantage of a silent profession. His calling is recent. Its work is not understood. He is judged only by results. The engineer who secures these results, as I have already pointed out, too often fails to get credit. Failing to get credit, he fails to have weight with the community, which blunders and wanders for lack of his guidance.

4 Who of you does not know of engineering blunders and engineering wastes because the large public of voters and the small public of directors are ignorant of what has been done in engineering? Both are ignorant because your profession does less than any other to educate the public. Look at the work for public health done by doctors, for new laws by lawyers. The engineer will never stand where he should in the state until he, too, discharges his duty at this point. Neither doctor nor lawyer speaks individually, nor need the engineer. Let local and national engineering societies speak.

5 Dr. Williams pointed out the field of legislation for the use of safety devices and suggested that the engineering society should guide and frame the laws. He spoke of the collapsing of a large concrete construction in Philadelphia, causing great loss of life, and pointed out that that was the time for the engineers, through the newspapers, to speak clearly, emphatically and authoritatively. Engineering societies should have their committees on legislation. The newspapers will always gladly print authoritative opinions. If failures in

construction or in design in public work had lucid authoritative exposition from engineering societies, if legislation were followed, if expert remedies were proposed by engineers for known evils, the public would come to look to these engineering societies for advice and the entire status and position of the engineer would change, and he would have the weight he should have as the maker of our current civilization, because he would be doing his duty in educating the great public.

HON. GEORGE W. GUTHRIE[1] urged that public evils should be pointed out and the remedy applied by men whose training fits them to deal with the questions, and whose character and abilities command the confidence of the people. If the people could understand from such men what was needed, how to do it and what it might cost, progress would be more rapid.

MR. FRED. W. TAYLOR said: Mr. Cooke's paper presents a large and almost unoccupied field of usefulness for our Society, and I trust that we may not be slow in acting upon his suggestions. What are we to do as a society towards coöperating with the public, that we are not now doing as individual engineers? It would seem on second thought that there must be a large range of subjects in which the Society either through committees or through a debating section, or through its employees (the Secretary and his executive staff) can coöperate with the public in a manner which would be entirely impossible for us as individual engineers.

2 Mechanical engineering problems are constantly recurring in the management and development of our cities, for example, which if not identical are yet similar, and it is to our Society, through its appropriate committees, that public officials should properly look for the standards, both mechanical and in method of procedure, which they should use in the solution of these problems. The standards recommended by the committees of our Society should have, and in fact have had, a weight and influence far beyond that of any individual engineer, however eminent. One of several illustrations of this will be found in the standard method of making boiler tests recommended by our committee, which has been for several years the standard practically accepted throughout this country, and is likely to remain so.

3 As we all know, our most able and public-spirited Secretary has gained the complete confidence of the Administration at Washington

[1] Hon. Geo. W. Guthrie, Mayor of Pittsburg.

by the efficient manner in which he helped the President in making a success of the White House Conference on the Conservation of our National Resources.

4 But few of our members, however, have heard of another instance in which our Society has been of great help to the Administration. Mr. Rice, this summer, and on very short notice, at the request of Dr. Rowe, chairman of the Delegation of the U. S. to the Pan-American Scientific Congress, secured 16 papers, some of them written by the most prominent engineers in their specialties, in this country, which are to be read at the First Pan-American Scientific Congress in Santiago, Chile. These papers should be of the greatest interest to our neighbors in South America, and should materially help in promoting friendly relations with them, and incidentally should direct their attention toward American engineers, and our standards and methods.

5 Der Verein deutscher Ingenieure, certainly the largest engineering society in the world, and in many respects the most successful, has established perhaps more intimate and useful relations with the people and the government than any other engineering association.

6 Their committee on education, for example, has proved itself of so great practical value to the German government and to the engineering and technical schools, that no important step in this educational field is taken without obtaining its advice.

7 Our Society has as yet appointed no committee on engineering education, although we have recently named two members on the Joint Committee for the Promotion of Engineering Education, appointed by the four large engineering societies and the American Society for the Promotion of Engineering Education.

Dr. Arthur T. Hadley[1] said he believed the engineering profession would not reach its highest position of influence until it had appreciated more fully the opportunities indicated in Mr. Cooke's paper.

Mr. Frank Miles Day[2] spoke of the work the American Institute of Architects has done in service to the public. It has fostered the movement for civic improvement both by the Institute proper and

[1] Arthur T. Hadley, LL.D., President of Yale University.

[2] Mr. Frank Miles Day, Architect, Philadelphia, Pa., lecturer on architecture at Harvard University.

by its local chapters in the larger cities. Among these is the Commission for the Improvement of Boston, which has now before it projects which for two years or more have received most careful study by the society.

2 In Cleveland, Ohio, the local chapter of the Institute gave the impetus that resulted in the appointment of an expert commission whose splendid plan for the grouping of public and semi-public buildings is now being carried into execution.

3 At the capital of the nation, the Institute was instrumental in securing the appointment of an expert commission, whose members served without remuneration, and formulated plans so convincing by the authority of their excellence, that Washington already seems a different city by virtue of work carried out in accordance with them.

4 Although in these and many other instances, the Institute has done notable work in assisting municipalities to solve difficult problems worthily, there is still a strong feeling among many of its members that the Institute is not closely enough in touch with the general public.

5 To improve this relation, the last convention directed that a Committee on Relations with the Public be appointed. Although there has not as yet been time for extended work on the part of this committee, its program includes:

a An attempt to secure, through the lay press, more worthy criticism of important buildings as they are from time to time completed, more adequate reports of the annual convention of the Institute, and more intelligent notices of architectural exhibitions.

b A series of magazine articles on the status and duties of the architect, on good and bad professional practice, on the evils of ill-regulated competition and on kindred subjects.

c An effort not only to interpret the aims and ideals of the profession to the public, but to assist the public in the conception that architecture is one of the fine arts.

6 It is hoped that something may also be done to impress upon the public the need of sound training for the architect, something that may help to deter the half-prepared or wholly unprepared youth from attempting a career that requires the fullest preparation, a preparation not merely of a highly special and technical character but a foundation of broad general culture equivalent to that indicated by the degree of Bachelor of Arts.

7 The Institute, from time to time, shows its interest in matters of import to the public by holding open meetings at which such subjects are discussed, or at which it evidences its interest in the sister arts of painting and sculpture by giving exhibitions of them.

8 The relations of technical, artistic and learned societies to each other are important. At present, these relations are by no means as close or profitable as they might be. To speak only of mechanical engineers and architects: it is fair to say that owing to the necessity for the services of engineers of high attainments in designing the mechanical and electrical equipment of modern buildings, architects and engineers have been brought in much closer touch than formerly and to their mutual advantage. But cannot the architects, as a body, through their Institute, be of service to the engineers as a body through their Society, and conversely, cannot the Society be of use to the Institute?

Mr. H. F. J. Porter thought that the conditions affecting the two classes, the employer and the employed, which compose "the people" to whom Mr. Cooke refers, should be improved. He maintained that some employers consider too lightly the social and economic effect of their methods on the people.

2 He contrasted the employer for whom "the best is none too good," and whose plant is in every respect a model, with the employer maintaining conditions so wretched as regards health and comfort and permitting the treatment of his help to be so bad that no self-respecting man would submit to them. It does not require a very vivid imagination to appreciate the difference in effects produced by these methods, either on the organizations or on the communities in which these plants are located. There is also another potent element affecting the relations between capital and labor in which "the people" are vitally interested. This is composed of the rich men who have acquired controlling interests in successful enterprises by advancing money on account of friends who have become involved. They are employers by accident and take no further interest in the affairs of the enterprise than to get returns on their investments. Their injunctions are to "keep down expenses." Such employers consider too lightly the social and economic effect of their methods on "the people."

3 Members of the Society engaged in reorganizing shops witness that a large proportion of failures in industrial enterprises are due to arbitrary methods of management. Changes in the methods of

management have become necessary and are directed away from the centralized or one-man source of authority towards distributed power or committee management. The executives of our largest industrial enterprises are urging their employees to become stock holders and thus part owners having a voice in the management.

4 Now that our Society is enlarging its scope and offering opportunities for the establishment of sections, to the membership of which others than those qualified as engineers are eligible, it seems to me that the opportunity is here presented for managers and those interested in management to get together for the interchange of views and the discussion of those questions which have so important a bearing on the industries of this country, and therefore upon "the people" at large who are directly affected by them.

Mr. Arthur L. Church described the methods by which the University of Pennsylvania keeps itself before the public in order that the public may take an intelligent interest in it. He further said:

2 I heartily agree with Mr. Cooke's suggestion that technical societies should take an active and altruistic interest in questions affecting the public weal. This has been done in many instances by the Franklin Institute of Philadelphia, and it accounts in some measure for the hold which that institution has on the public and the interest which the public takes in its welfare.

Prof. A. W. Moseley thought that an intelligent and unselfish meeting of the opportunities of the engineer would bring him before the public in a more evidently useful and generally prominent light. As an instance he suggests that the introduction of safety appliances be urged universally by members of the Society. He also recommends to the attention the æsthetic as well as the ethical side of engineering, quoting from Dr. Waddell's De Pontibus: "In all structural work the subject of æsthetics must be duly considered: and all designs are to be made in harmony with the principles thereof, to as great an extent as the money available for the work will permit, or the environment of the structure calls for."

Mr. James M. Dodge heartily approved of the suggestions in Mr. Cooke's paper and expressed himself as believing that the ball which Mr. Cooke had set rolling would be accelerated in its progress by the professions for all time to come.

Mr. Ambrose Swasey said that the field of the engineer is broadening all the time, and that this paper contemplates a yet larger field of usefulness. He quoted Congressman Burton as saying in connection with the conservation movement: "By all means the engineer should be found in the front rank, for he is a most important factor in this splendid work which we are about to undertake, a work of which we have just reached the edge, and which will go on and on, increasing as the years progress."

2 Mr. Swasey further said: What Mr. Cooke has said means a step in advance for this Society and for the engineers of this country. It means that we are going to pay more attention to public matters, not simply to our private interests, but to the good of the people as a whole. In this connection I wish to present the following resolution:

3 Resolved, that we recommend to the Council the appointment of a Professional Committee, to investigate, consider and report on the methods whereby the Society may more directly coöperate with the public on engineering matters, and on the general policy which should control such coöperation.

Mr. Charles Wallace Hunt said: This resolution, which Mr. Swasey has proposed, is really broadening the work of the Society. It may become a national engineering and economic movement, which may develop later as a section of the Society. With that in view, I second the motion made by Mr. Swasey. [The resolution was also seconded by Mr. Fred. W. Taylor and unanimously adopted.—Editor.]

Prof. F. R. Hutton called attention to the necessity for an amendment to the Constitution for this purpose, and gave notice thereupon to amend Article C45 at the Spring Meeting of the Society, at which time such amendment would be brought up for discussion.

Mr. Oberlin Smith agreed in general with Mr. Cooke's idea of a standing committee upon relations with the public, with the additional suggestion that such a committee might also act as a committee upon a Code of Ethics, a matter which other societies are taking up and which we cannot afford to neglect. He said, it is true that engineers as a body live too much to themselves and for their work and do not sufficiently affiliate with the public in general.

2 He urged the higher self-cultivation of all engineers and regretted that with the exception of West Point and Annapolis, few of our colleges and technical schools give enough attention to this

matter. He suggested that it would be a wise policy for this Society to urge upon the technical schools the adoption of broader and more liberal schemes of education, especially during the students' early years.

3 He stated that the public had not yet fully discriminated between the machinist who shapes the iron into a steam engine, or the engineman who oils and cleans and watches it run, and the professional engineer, who with the necessary ability, education and experience, has designed it. He hoped that the nomenclature would be improved, and that we would either drop the name of engineer or try in earnest to limit it to professional men, using other names for the lower grades. He said further: Knowing what our profession is and does, it seems remarkable that it is not more often represented in Congress, in the Cabinet and even in the White House.

The Author The engineering profession may almost be said to accept the conception of its duty toward the public as outlined in this paper, since all the discussion before the Society, in the technical press and elsewhere, has apparently been favorable. Even those who place the highest value on the work of the engineer are willing to admit that he has failed in this public function. There is little therefore to be gained from further discussion. Carlyle has said that "The end of Life is an Action, not a Thought, though it were the noblest." This injunction comes with added weight to a profession that stands for action, not speculation.

2 It is not too much to expect that The American Society of Mechanical Engineers should lead in the work of transforming this thought into action. Already its committees are showing in their various plannings the quickening thrill of a broader vision. It is nevertheless true that the Society acting as a corporate entity can be effective in this work only as the individual activities of its members make it possible and as it is held up to the work by what may be called the suggestive influence of individual members.

3 Is there not the danger that our progress may be retarded by our holding back as a Society and as individual members in order to take part in some great work which is to benefit the public in a large way? Doubtless there are such undertakings ahead of us. But one small piece of engineering work of a purely public character done by an engineer or an association of engineers with the utmost efficiency and done at once will advance the whole program more than would the proposing of a dozen more ambitious schemes which might one by one die in the process of being discussed.

No. 1210

THE PRESENT STATUS OF MILITARY AËRONAUTICS

By George O. Squier, Ph.D, Washington, D. C.
Major, Signal Corps, U. S. Army
Non-Member

It is a matter of first significance that The American Society of Mechanical Engineers, composed of a body of highly trained and serious minded men, should be considering in annual meeting assembled the subject of aërial navigation. Five years ago such a subject could scarcely have had a place on the list of professional papers on your program. The present period will ever be memorable in the history of the world for the first public demonstrations of the practicability of mechanical flight. In fact, at the present moment a resistless wave of enthusiasm and endeavor, sweeping away every prejudice, is passing over the entire civilized world, fixing the attention of all classes upon the problem of flight. France, Germany, and England are in a state of frenzied interest in this subject, and each period of a single month sees some new step accomplished in the march of progress. The Universal Highway is at last to be made available for the uses of mankind, with its consequent influence upon our modes of life and thought.

2 The subject of war balloons and their accessories pertains by law to the Signal Corps of the Army, and some months since an invitation was extended to the Chief Signal Officer of the Army, Genl. James Allen, to meet with you on this occasion and present to this distinguished body of practical engineers an outline of the work of the Government in this direction. On account of pressure of official duties General Allen has designated me to perform this duty, and notwithstanding a keen consciousness of personal shortcomings, yet I would be indeed lacking in sentiment if I failed to acknowledge the honor felt in appearing here today to present such

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

a subject for the first time before a national body of American engineers.

3 At the outset, it must be stated that the subject is so vast in its scientific details and that data and results are being obtained so rapidly that it is manifestly impossible to present more than the merest outline of the present state of this new Science and Art within the limits of a short paper. From the earliest times men have dreamed of imitating the birds in sailing through the air, yet it is only within a very few years that the strength of materials and the mechanical construction of motors have reached a state to make power-flight possible. The industrial development of the automobile has been a powerful ally in the realization of mechanical flight, and the engineering profession finds itself equipped and ready to further the development of this great problem.

4 On December 23, 1907, the Signal Corps of the Army issued a public advertisement and specification calling for bids for furnishing the Government with a heavier-than-air flying machine. A copy of this specification is appended to this paper as of possible historical interest.

5 The conditions of this specification require that the Government be furnished with a heavier-than-air flying machine capable of carrying one passenger besides the aviator, and it must remain in the air on an endurance test for a period of one hour without landing, and must also be subjected to a speed test over a measured course of more than five miles, against and with the wind, attaining a minimum speed of 36 miles per hour. The machine must, in addition, carry fuel for a continuous flight of not less than 125 miles.

6 In preparing this specification, it was purposely sought to leave the bidder perfectly free in the methods to be employed, and he was not restricted as to type or design. At the time this specification was issued, eleven months ago, the conditions were publicly regarded as being unusually severe and far beyond the state of the art at that time. That these conditions were justified has been subsequently proved, as is now well known.

7 Although the public advertisement called for but one heavier-than-air machine, yet when the bids were opened it was found possible, through the coöperation of the Board of Ordnance and Fortification, to award a contract to each bidder who complied with the requirements of the law in every respect, and consequently contracts were ultimately awarded to the Wright Brothers of Dayton, Ohio, for the

sum of $25 000 for a 40-mile speed, and also to A. M. Herring of New York, for the sum of $20 000.

8 It was believed that the acceptance by the Government of each of the bids submitted instead of but one of them would serve as an additional stimulus to develop practical aviation in the United States, and at the same time serve to supply the War Department with machines needed in military service. This dual object,—to advance a new art of interest to the nation as a whole, and to secure necessary equipment for the military establishment,—has been in the past and is at present the policy of the Signal Corps of the Army.

9 The result of issuing this specification, as well as a similar one for supplying a small dirigible balloon for the preliminary training of the men of the Signal Corps, was an awakening of interest in this subject throughout the country to such an extent that the Signal Office continues to receive daily a large number of letters, plans, and models proposing manifold schemes for navigating the air.

10 The Aëronautical Division of the Office of the Chief Signal Officer of the Army was organized on July 1, 1907, and the Aëronautical Board of the Signal Corps was appointed in July of the current year for conducting tests of dirigible balloons and aëroplanes under existing contracts.

11 It should be stated that the mention of particular types of dirigible balloons and aëroplanes in this paper must not be considered as an official indorsement of these particular machines, nor the failure to mention other types be construed to indicate a lack of equal recognition of the merits of the latter. In the case of the Wright Brothers, however, it is desired to associate the Signal Corps of the Army publicly and officially with the present universal recognition of their work in advancing the science and art of aviation. These results have been due to the persistence, daring, and intelligence of these American gentlemen, to whom the whole world is now paying homage. It will ever be recorded that the classic series of public demonstrations first made by Orville Wright at the government testing grounds at Fort Myer, Va., in September, 1908, and by Wilbur Wright at Le Mans, France, made a profound impression throughout the world, and kindled especially the patriotic spirit of the American people.

12 There are two general classes of vehicles of the air, (*a*) those which depend for their support upon the buoyancy of some gas lighter than air, and (*b*) those which depend for such support upon the dynamic reaction of the air itself. These classes are designated

a Lighter-than-air types:
Free balloons, dirigible balloons or airships
b Heavier-than-air types:
Aëroplanes, orthopters, helicopters, etc.

13 It should be remarked, however, that these two general classes exhibit a growing tendency to overlap each other. For example, the latest dirigible balloons are partly operated by means of aëroplane surfaces, and are also often balanced so as to be slightly heavier than the air in which they move, employing the propeller thrust and rudder surfaces to control the altitude.

AËROSTATION

14 Captive and free balloons, with the necessary apparatus and devices for operating the same, have been for many years considered an essential part of the military establishment of every first-class Power. They played a conspicuous part in the siege of Paris, and were often valuable in our own Civil War. The construction and operation of aërostats are too well understood to need further attention here.

Successful Military Dirigible Balloons

FRANCE

15 Two types of dirigible balloons have been used in the French Army; first the *Patrie,* and second the *Ville de Paris.*

16 The *Patrie* was developed by Julliot, an engineer employed by the Lebaudy Brothers at their sugar refinery in Paris. A history of his work beginning in 1896 is fully given in *La Conquête de l'Air.*

THE PATRIE

17 The *Patrie,* the third of its type, was first operated in 1906. The gas bag of the first balloon was built by Surcouf at Billancourt, Paris. The mechanical part was built at the Lebaudy Sugar Refinery. Since then the gas bags have been built at the Lebaudy balloon shed at Moisson, near Paris, under the direction of their aëronaut, Juchmés. The gas bag of the *Patrie* was 197 ft. long with a maximum diameter of 33 ft. 9 in., situated about 2/5 of the length from the front; volume 111 250 cu. ft.; length approximately six diameters. This relation, together with the cigar shape, is in accordance with the plans of Colonel

Renard's dirigible, built and operated in France in 1884; the same general shape and proportions being found in the *Ville de Paris.*

18 The first Lebaudy was pointed at the rear, which is generally admitted to be the proper shape for the least resistance, but to maintain stability it was found necessary to put a horizontal and vertical plane there, so that it had to be made an ellipsoid of revolution to give attachment for these planes.

19 The ballonet for air had a capacity of 22 958 cu. ft. or about $\frac{1}{5}$ of the total volume. This is calculated to permit reaching a height of about one mile and returning to the earth, keeping the gas bag always rigid. To descend from a height of one mile, gas would be released by the valve, then air pumped into the ballonet to keep the gas bag rigid, these two operations being carried on alternately. On reaching the ground from the height of one mile the air would be at the middle of the lower part of the gas bag and would not entirely fill the ballonet. To prevent the air from rolling from one end to the other when the airship pitches, thus producing instability, the ballonet was divided into three compartments by impermeable cloth partitions. Numerous small holes were pierced in these partitions through which the air finally reached the two end compartments.

20 In September 1907, the *Patrie* was enlarged by 17 660 cu. ft. by the addition of a cylindrical section at the maximum diameter, increasing the length but not the maximum diameter.

21 The *gas bag* is cut in panels; the material is a rubber cloth made by the Continental Tire Company at Hanover, Germany. It consists of four layers arranged as follows:

	Weight oz. per square yard
a Outer layer of cotton cloth covered with lead chromate	2.5
b Layer of vulcanized rubber	2.5
c Layer of cotton cloth	2.5
d Inner layer of vulcanized rubber	2.21
Total weight	9.71

22 A strip of this cloth one foot wide tears at a tension of about 934 lb. A pressure of about 1 in. of water can be maintained in the gas bag without danger. The lead chromate on the outside is to prevent the entrance of the actinic rays of the sun which would cause the rubber to deteriorate. The heavy layer of rubber is to prevent the

leaking of the gas. The inner layer of rubber is merely to prevent deterioration of the cloth by impurities in the gas. This material has the warp of the two layers of cotton cloth running in the same direction and is called straight thread. The material in the ballonet weighs only about 7¾ oz. per sq. yd. and has a strength of about 336 lb. per running foot. When the *Patrie* was enlarged in September 1907, the specifications for the material allowed a maximum weight of 10 oz. per sq. yd., a minimum strength of 907 lb. per running foot, and a loss of 5.1 cu. in. of hydrogen per square yard in twenty-four hours at a pressure of 1.18 in. of water. Bands of cloth are pasted over the seams inside and out with a solution of rubber to prevent leaking through the stitches.

23 *Suspension.* One of the characteristics of the *Patrie* is the "short" suspension. The weight of the car is distributed over only about 70 ft. of the length of the gas bag. To do this, an elliptical shaped frame of nickel steel tubes is attached to the bottom of the gas bag; steel cables run from this down to the car. A small hemp net is attached to the gas bag by means of short wooden cross pieces or toggles which are let into holes in a strong canvas band sewed directly on the gas bag. The metal frame, or platform, is attached to this net by means of toggles, so that it can be quickly removed in dismounting the airship for transportation. The frame can also be taken apart. Twenty-eight steel cables about 0.2 in. in diameter run from the frame down to the car, and are arranged in triangles. Due to the impossibility of deforming a triangle, rigidity is maintained between the car and gas bag.

24 The objection to the "short" suspension of the *Patrie* is the deformation of the gas bag. A distinct curve can be seen in the middle.

25 *The Car.* The car is made of nickel steel tubes (12 per cent nickel). This metal gives the greatest strength for minimum weight. The car is boat-shaped, about 16 ft. long, about 5 ft. wide, and 2½ ft. high. About 11 ft. separates the car from the gas bag. To prevent any chance of the fire from the engine communicating with the hydrogen, the steel framework under the gas bag is covered with a non-combustible material.

26 The pilot stands at the front of the car, the engine is in the middle, the engineer at the rear. Provision is made for mounting a telephotographic apparatus, and for a 100-candle-power acetylene search-light. A strong pyramidal structure of steel is built under the car, pointing downward. In landing the point comes to the ground

first and this protects the car, and especially the propellers, from being damaged. The car is covered to reduce air resistance. It is so low, however, that part of the equipment and most of the bodies of those inside are exposed, so that the total resistance of the car is large.

27 *The Motor.* The first Lebaudy had a 40 h. p. Daimler-Mercedes benzine motor. The *Patrie* was driven by a 60 to 70 h.p., 4-cylinder Panhard Levassor benzine motor, making 1000 r.p.m.

28 *The Propellers.* There are two steel propellers 8½ ft. in diameter (two blades each) placed at each side of the engine, thus giving the shortest and most economical transmission. To avoid any tendency to twist the car, the propellers turn in opposite directions. They are "high speed," making 1000 to 1200 r.p.m.

29 The gasolene tank is placed under the car inside the pyramidal frame. The gasolene is forced up to the motor by air compression. The exhaust is under the rear of the car pointing down and is covered with a metal gauze to prevent flames coming out. The fan which drives the air into the ballonet is run by the motor, but a dynamo is also provided so that the fan can always be kept running even if the motor stops. This is very essential as the pressure must be maintained inside the gas bag so that the latter will remain rigid and keep its form. There are five valves in all, part automatic and part both automatic and also controlled from the car with cords. The valves in the ballonet open automatically at less pressure than the gas valves, so that when the gas expands all the air is driven out of the ballonet before there is any loss of gas. The ballonet valves open at a pressure of about 0.78 in. of water, the gas valves at about 2 in.

30 *Stability.* Vertical stability is maintained by means of fixed horizontal planes. One having a surface of 150 sq. ft. is attached at the rear of the gas bag, and due to its distance from the center of gravity is very efficient. The elliptical frame attached under the gas bag has an area of 1055 sq. ft. but due to its proximity to the center of gravity, has little effect on the stability. Just behind the elliptical frame is an arrangement similar to the feathering on an arrow. It consists of a horizontal plane of 150 sq. ft., and a vertical plane of 113 sq. ft. To maintain horizontal stability, that is, to enable the airship to move forward in a straight line without veering to the sides, fixed vertical planes are used. One runs from the center to the rear of the elliptical frame and has an area of 108 sq. ft.

31 In addition to the vertical surface of 113 sq. ft. at the rear of the elliptical frame, there is a fixed plane of 150 sq. ft. at the rear of

the gas bag. To fasten the two perpendicular planes at the rear of the gas bag, cloth flaps are sewed directly on the gas bag. Nickel steel tubes are placed in the flaps which are then laced over the tubes. With these tubes as a base, a light tube and wire framework is attached and water-proof cloth laced on this framework. Additional braces run from one surface to the other and from each surface to the gas bag. The rudder is at the rear under the gas bag. It has about 150 sq. ft. and is balanced.

32 A movable horizontal plane near the center of gravity, above the car, is used to produce rising or descending motion, or to prevent an involuntary rising or falling of the airship due to expansion or contraction of the gas or to other causes. After the adoption of this movable horizontal plane, the loss of gas and ballast was reduced to a minimum. Ballast is carried in 10 and 20 lb. sand bags. A pipe runs through the bottom of the car from which the ballast is thrown.

33 There are two long guide ropes, one attached at the front of the elliptical frame and the other on the car. On landing, the one in front is seized first so as to hold the airship with the head to the wind. The motor may then be stopped and the descent made by pulling down on both guide ropes. A heavy rope, 22 ft. long, weighing 110 lb., is attached on the end of a 164 ft. guide rope. This can be dropped out on landing to prevent coming to the ground too rapidly. The equipment of the car includes a "siren," speaking trumpet, carrier pigeons, iron pins and a rope for anchoring the airship, reserve supply of fuel and water, and fire extinguisher.

34 After being enlarged in September 1907 the *Patrie* made a number of long trips at an altitude of 2500 to 3000 ft. In November 1907, she went from Paris to Verdun, near the German frontier, a distance of about 175 miles, in about 7 hours, carrying four persons. This trip was made in a light wind blowing from the northeast. Her course was east, so that the wind was unfavorable. On Friday, November 29, 1907, during a flight near Verdun, the motor stopped due to difficulty with the carburetter. The airship drifted with the wind to a village about 10 miles away where she was safely landed. The carburetter was repaired on the 30th. Soon after, a strong wind came up and tore loose some of the iron pickets with which the airship was anchored. This allowed it to swing broadside to the wind; it then tilted over on the side far enough to let some of the ballast bags fall out. The 150 or 200 soldiers who were holding the ropes were pulled along the ground until directed by the officer in charge to let go. After being released, the airship rose and was carried by the wind

across the north of France, the English Channel, and into the north of Ireland. It struck the earth there, breaking off one of the propellers, and then drifted out to sea.

THE REPUBLIQUE

35 This is the latest of the French military dirigible balloons, and differs but slightly from its predecessor, the *Patrie.* The volume has been increased by about 2000 cu. ft. The length has been reduced to 200 ft. and the maximum diameter increased to 35½ ft. The shape of the gas bag accounts for the 2000 additional cubic feet of volume. The motor and propellers are as in the *Patrie.* The total lifting capacity is 9000 lb., of which 2700 lb. are available for passengers, fuel, ballast, instruments, etc. Its best performance was a 125-mile flight made in 6½ hours against an unfavorable wind.

36 The material for the gas bag of the new airship was furnished by the Continental Tire Company. It is made up as follows:

	Weight oz. per square yard
Outer yellow cotton layer	3.25
Layer of vulcanized rubber	3.25
Layer of cotton cloth	3.25
Inner layer of rubber	0.73
Total weight	10.48

37 It is interesting to note the changes which this type has undergone since the first one was built. The *Jaune,* constructed in 1902–1903, was pointed at the rear and had no stability plane there; later it was rounded off at the rear and a fixed horizontal plane attached. Finally a fixed vertical plane was added. The gas bag has been increased in capacity from 80 670 to about 131 000 cu. ft. The manufacturers have been able to increase the strength of the material of which the gas bag is made, without materially increasing the weight. The rudder has been altered somewhat in form. It was first pivoted on its front edge, but later on a vertical axis, somewhat to the rear of this edge. With the increase in size, has come an increase in carrying capacity and consequently a greater speed and more widely extended field of action.

THE VILLE DE PARIS

38 This airship was constructed for Deutsch de la Meurthe, of Paris, who has done a great deal to encourage aërial navigation. The

first *Ville de Paris* was built in 1902, on plans drawn by Tatin, a French aëronautical engineer. It was not a success. Its successor was built in 1906, on plans of Surcouf, an aëronautical engineer and balloon builder. The gas bag was built at his works in Billancourt, the mechanical part at the Voisin shop, also in Billancourt. The plans are based on those of Colonel Renard's airship, the *France*, built in 1884, and the *Ville de Paris* resembles the older airship in many particulars. In September 1907, Mr. Deutsch offered the use of his airship to the French Government. The offer was accepted, but delivery was not to be made except in case of war or emergency. When the *Patrie* was lost in November 1907, the military authorities immediately took over the Deutsch airship.

39 *Gas Bag.* The gas bag is 200 ft. long for a maximum diameter of 34½ ft., giving a length of about 6 diameters, as in the *France* and the *Patrie*. Volume 112 847 cu. ft., maximum diameter at about ⅜ of the distance from the front, approximately, as in the *Patrie*. The middle section is cylindrical with conical sections in front and rear. At the extreme rear is a cylindrical section with eight smaller cylinders attached to it. The ballonet has a volume of 21 192 cu. ft., or about ⅕ of the whole volume, the same proportion found in the *Patrie*. The ballonet is divided into three compartments from front to rear. The division walls are of permeable cloth, and are not fastened to the bottom so that when the middle compartment fills with air, and the ballonet rises, the division walls are lifted up from the bottom of the gas bag, and there is free communication between the three compartments. The gas bagis madeup of a series of strips perpendicular to a meridian line. These strips run around the bag, their ends meeting on the under meridian. This is known as the "brachistode" method of cutting out the material, and has the advantage of bringing the seams parallel to the line of greatest tension. They are therefore more likely to remain tight and not allow the escape of gas. The disadvantage lies in the fact that there is a loss of 33⅓ per cent of material in cutting. The material was furnished by the Continental Tire Company, and has approximately the same tensile strength and weight as that used in the *Patrie*. It differs from the other in one important feature—it is diagonal-thread, that is, the warp of the outer layer of cotton cloth makes an angle of 45 deg. with the warp of the inner layer of cotton cloth. The result is to localize a rip or tear in the material. A tear in the straight thread material will continue along the warp, or the weave, until it reaches a seam.

40 *Valves.* There are five in all, made of steel, about fourteen inches in diameter; one on the top connected to the car by a cord, operated by hand only; two near the rear underneath. These are automatic but can be operated by hand from the car. Two ballonet valves directly under the middle are automatic and are also operated from the car by hand. The ballonet valves open automatically at a pressure of $\frac{2}{3}$ in. of water, the gas valves open at a higher pressure.

41 *Suspension.* This airship has the "long" suspension. That is, the weight is distributed along practically the entire length of the gas bag. A doubled band of heavy canvas is sewn with six rows of stitches along the side of the gas bag. Hemp ropes running into steel cables transmit most of the weight of the car to these two canvas bands and thus to the gas bag. On both sides and below these first bands are two more. Lines run from these to points half way between the gas bag and the car, then radiate from these points to different points of attachment on the car. This gives the triangular or non-deformable system of suspension, which is necessary in order to have the car and gas bag rigidly attached to each other. With this "long" suspension, the *Ville de Paris* does not have the deformation so noticeable in the gas bag of the *Patrie.*

42 *The Car.* This is in the form of a trestle. It is built of wood with aluminum joints and 0.12 in. wire tension members. It is 115 ft. long, nearly 7 ft. high at the middle, and a little over 5½ ft. wide at the middle. It weighs 660 lb. and is considered unnecessarily large and heavy. The engine and engineer are well to the front, the aëronaut with steering wheels is about at the center of gravity.

43 *Motor.* The motor is a 70 to 75 h.p. "Argus," and is exceptionally heavy.

44 *Propeller.* The propeller is placed at the front end of the car. It thus has the advantage of working in undisturbed air; the disadvantage is the long transmission and difficulty in attaching the propeller rigidly. It has two blades and is 19.68 ft. long with a pitch of 26.24 ft. The blades are of cedar with a steel arm. The propeller makes a maximum of 250 turns per minute when the engine is making 900 rev. Its great diameter and width compensate for its small speed.

45 *Stability.* This is maintained entirely by the cylinders at the rear. Counting the larger one to which the smaller ones are attached, there are five, arranged side by side corresponding to the horizontal planes of the *Patrie,* and five vertical ones corresponding to the *Patrie's* vertical planes. The volume of the small cylinders is so calculated

that the gas in them is just sufficient to lift their weight, so they neither increase or decrease the ascensional force of the whole. The horizontal projection of these cylinders is 1076 sq. ft. The center of this projection is 72 ft. from the center of gravity of the gas. The great objection to this method of obtaining stability, is the air resistance due to these cylinders, and the consequent loss of speed. The stability of the *Ville de Paris* in a vertical plane is said to be superior to that of the *Patrie*, due to the fact that the stability planes of the latter do not always remain rigid. The independent velocity of the *Ville de Paris* probably never exceeded 25 miles an hour.

46 *The Rudder.* The rudder has a double surface of 150 sq. ft. placed at the rear end of the car, 72 ft. from the center of gravity. It is not balanced, but is inclined slightly to the rear so that its weight would make it point directly to the rear if the steering gear should break. Two pairs of movable horizontal planes, one at the rear of the car having 43 sq. ft., and one at the center of gravity (as on the *Patrie*) having 86 sq. ft., serve to drive the airship up or down without losing gas or ballast.

47 *Guide Ropes.* A 400 ft. guide rope is attached at the front end of the car. A 230 ft. guide rope is attached to the car at the center of gravity.

48 About thirty men are required to maneuver the *Ville de Paris* on the ground. The pilot has three steering wheels, one for the rudder and two for the movable horizontal planes. The instruments used are an aneroid barometer, a registering barometer giving heights up to 1600 ft. and an ordinary dynamometer which can be connected either with the gas bag or ballonet by turning a valve. A double column of water is also connected to the tube to act as a check on the dynamometer. Due to the vibration of the car caused by the motor, these instruments are suspended by rubber attachments. Even with this arrangement, it is necessary to steady the aneroid barometer with the hand in order to read it. The vibration prevents the use of the statoscope.

ENGLAND

MILITARY DIRIGIBLE NO. 1

49 The gas bag of this airship was built about five years ago by Colonel Templar, formerly in command of the aëronautical establishment at Aldershot. His successor, Colonel Capper, built the mechanical part during the spring and summer of 1907, with the assistance of Mr.

S. F. Cody, a mechanical engineer. It was operated by Colonel Capper as pilot, with Mr. Cody in charge of the engine. Several ascents were made at Aldershot. In October 1907, they made a trip from Aldershot to London, a distance of about 40 miles, landing at the Crystal Palace. For several days the rain and wind prevented attempting the return journey. On October 10 a strong wind threatened to carry away the airship, so the gas bag was cut open by the sergeant in charge.

50 *Gas Bag.* This is made of eight layers of gold beater's skin. It is cylindrical in shape with spherical ends. Volume 84 768 cu. ft.; length 111½ ft.; maximum diameter, 31½ ft. The elongation therefore is only about 3⅗. There is no ballonet, but due to the toughness of the gold beater's skin, a much higher pressure can safely be maintained than in gas bags of rubber cloth. Without a ballonet, however, it would not be safe to rise to the heights reached by the *Patrie*.

51 *Valves.* The valves are made of aluminum and are about 12 in. in diameter.

52 *Suspension.* In this airship they have succeeded in obtaining a "long" suspension with a short boat-shaped car, a combination very much to be desired, as it distributes the weight over the entire length of the gas bag and gives the best form of car for purposes of observation and for maneuvering on the ground. To obtain this combination they have had to construct a very heavy steel framework which cuts down materially the carrying capacity, and moreover, this framework adds greatly to the air resistance. This is the only airship in Europe having a network to support the car. In addition, four silk bands are passed over the gas bag and wires run from their extremities down to the steel frame. This steel frame is in two tiers; the upper is rectangular in cross-section and supports the rudder and planes; the lower part is triangular in cross-section and supports the car. The joints are aluminum.

53 *The Car.* This is of steel and is about thirty feet long. To reduce air resistance, the car is covered with cloth.

54 *Motor.* A 40 to 50 h.p. 8-cylinder Antoinette motor is used. It is set up on top of the car. The benzine tanks are supported above in the framework. Gravity feed is used.

55 *Propellers.* There are two propellers, one on each side, with two blades each, as in the *Patrie*. They are made of aluminum, 10 ft. in diameter, and make 700 r.p.m. The transmission is by belt.

56 *Stability.* This is maintained by means of planes. At the extreme rear is a large fixed horizontal plane. In front of this is a pair of

hinged horizontal planes. Under this is the hexagonal shaped rudder. It is balanced. Two pairs of movable horizontal planes, 8 ft. by 4 ft., each placed at the front, serve to guide the airship up and down, as in the *Patrie* and *Ville de Paris.* These planes have additional inclined surfaces which are intended to increase the stability in a vertical plane. All these planes, both fixed and movable, are constructed like kites, of silk stretched on bamboo frames. The guide rope is 150 ft. long. Speed attained, about 16 miles per hour. This airship with a few improvements added has been in operation the past few months. The steel framework connecting the gas bag to the car is now entirely covered with canvas, which must reduce the resistance of the air very materially. The canvas covering, enclosing the entire bag, serves as a reinforcement to the latter and at the same time gives attachment to the suspension underneath. It is reported that a speed of 20 miles an hour has been attained with the reconstructed airship.

57 A pyramidal construction similar to that on the *Patrie* has been built under the center of the car to protect the car and propellers on landing. A single movable horizontal plane placed at the front end of the car and operated by the pilot, controls the vertical motion.

GERMANY

58 Three different types of airships are being developed in Germany. The *Gross* is the design of Major Von Gross, who commands the Balloon Battalion at Tegel near Berlin. The *Parseval* is being developed by Major Von Parseval, a retired German Officer, and the *Zeppelin* is the design of Count Zeppelin, also a retired officer of the German Army.

THE GROSS

59 The first airship of this type made its first ascension on July 23, 1907. The mechanical part was built at Siemen's Electrical Works in Berlin; the gas bag by the Riedinger firm in Augsburg.

60 *Gas Bag.* The gas bag is made of rubber cloth furnished by the Continental Tire Company, similar to that used in the *Ville de Paris.* It is diagonal-thread, but there is no inner layer of rubber, as they do not fear damage from impurities in the hydrogen gas. Length, 131¼ ft.; maximum diameter, about 39⅓ ft.; volume 63 576 cu. ft.; elongation about 3⅓. The form cylindrical with spherical cones at the ends, the whole being symmetrical.

61 *Suspension.* The suspension is practically the same as that of the *Patrie.* A steel and aluminum frame is attached to the lower part of the gas bag, and the car is suspended on this by steel cables. The objection to this system is even more apparent in the *Gross* than in the *Patrie.* A marked dip along the upper meridian of the gas bag shows plainly the deformation.

62 *The Car.* The car is boat-shaped like that of the *Patrie.* It is suspended thirteen feet below the gas bag.

63 *Motor.* The motor is a 20 to 24 h.p., 4-cylinder "Daimler-Mercedes."

64 *Propellers.* There are two propellers $8\frac{2}{10}$ ft. in diameter, each having two blades. They are placed one on each side, but well up under the gas bag near the center of resistance. The transmission is by belt. The propellers make 800 r.p.m.

65 *Stability.* The same system, with planes, is used in the *Von Gross* as in the *Patrie,* but it is not nearly so well developed. At the rear of the rigid frame attached to the gas bag, are two fixed horizontal planes, one on each side. A fixed vertical plane runs down from between these horizontal planes, and is terminated at the rear by the rudder. A fixed horizontal plane is attached on the rear of the gas bag as in the *Patrie.* The method of attachment is the same, but the plane is put on before inflation in the Gross airship, afterwards in the *Patrie.* The stability of the Gross airship in a vertical plane is reported to be very good, but it is said to veer considerably in attempting to steer a straight course.

66 The many points of resemblance between this dirigible and the Lebaudy type are worthy of notice. The suspension or means of maintaining stability, and the disposition for driving, are in general the same. As first built, the *Gross* had a volume of 14 128 cu. ft. less than at present, and there was no horizontal plane at the rear of the gas bag. Its maximum speed is probably fifteen miles per hour. As a result of his experiments of 1907, Major Von Gross has this year produced a perfected airship built on the same lines as his first, but with greatly increased volume and dimensions. The latest one has a volume of 176 000 cu. ft., is driven by two 75 h.p. Daimler motors, and has a speed of 27 miles per hour.

67 On September 11, 1908, the Gross airship left Berlin at 10.25 p.m., carrying four passengers, and returned the next day at 11.30 a.m., having covered 176 miles in the period of a little over 13 hours. This is the longest trip, both in points of time and distance, ever made by any airship returning to the starting point.

THE PARSEVAL

68 The Parseval airship is owned and controlled by the Society for the Study of Motor Balloons. This organization, composed of capitalists, was formed practically at the command of the Emperor who is very much interested in aërial navigation. The society has a capital of 1 000 000 marks, owns the Parseval patents and is ready to construct airships of the Parseval type. The present airship was constructed by the Riedinger firm at Augsburg, and is operated from the balloon house of this society at Tegel, adjoining the military balloon house.

69 The gas bag is similar in construction to that of the "*Drachen*" balloon, used by the army for captive work. Volume, 113 000 cu. ft., length 190 ft., maximum diameter 30½ ft. It is cylindrical in shape, rounded at the front end and pointed at the rear. The material was furnished by the Continental Tire Company. It is diagonal-thread, weighing about $11\frac{2}{10}$ oz. per sq. yd. and having a strength of about 940 lb. per running foot. Its inner surface is covered with a layer of rubber.

70 *Ballonets.* There are two ballonets, one at each end, each having a capacity of 10 596 cu. ft. The material in the ballonet weighs about 8¼ oz. per sq. yd., the cotton layers being lighter than in the material for the gas bag. Air is pumped into the rear ballonet before leaving the ground, so that the airship operates with the front end inclined upward. The air striking underneath exerts an upward pressure, as on an aëroplane, and thus adds to its lifting capacity. Air is pumped into the ballonets from a fan operated by the motor. A complex valve just under the middle of the gas bag enables the engineer to drive air into either, or both ballonets. The valves also act automatically and release air from the ballonets at a pressure of about 0.9 in. of water.

71 In the middle of the top of the gas bag, is a valve for releas-the gas. It can be operated from the car, and opens automatically at a pressure of about 2 in. of water. Near the two ends and on opposite sides, are two rip strips controlled from the car by cords.

72 *Suspension.* The suspension is one of the characteristics of the airship and is protected by patents. The car has four trolleys, two on each side, which run on two steel cables. The car can run backwards and forwards on these cables, thus changing its position with relation to the gas bag. This is called "loose" suspension. Its object is to allow the car to take up, automatically, variations in

thrust due to the motor, and variations in resistance due to the air. Ramifications of hemp rope from these steel cables are sewn to a canvas strip which in turn is sewn to the gas bag. This part of the suspension is the same as in the Drachen balloon. The weight is distributed over the entire length of the gas bag.

73 *The Car.* The car is 16.4 ft. long and is built of steel tubes and wire. It is large enough to hold the motor and three men, though four or five may be taken.

74 *Motor.* The motor is a 110 h.p. Daimler-Mercedes. Sufficient gasolene is carried for a run of 12 hours.

75 *Propeller.* The propeller, like the suspension, is peculiar to this airship and is protected by patents. It has four cloth blades which hang limp when not turning. When the motor is running, these blades, which are carefully weighted with lead at certain points, assume the proper position due to the various forces acting. The diameter is 13¾ ft. The propeller is placed above the rear of the car near the center of resistance. Shaft transmission is used. The propeller makes 500 r.p.m. to 1000 of the motor. There is a space of 6½ ft. from the propeller blades to the gas bag, the bottom of the car being about 30 ft. from the gas bag. This propeller has the advantage of being very light. Its position, so far from the engine, necessarily incurs a great loss of power in transmission.

76 The steering wheel, at the front of the car, has a spring device for locking it in any position.

77 The 1908 model of this airship was constructed for the purpose of selling it to the Government. Among other requirements is a 12 hour flight without landing, and a sufficient speed to maneuver against a 22-mile wind. A third and larger airship of this type is now under construction.

THE ZEPPELIN

78 The Zeppelin airship, of which there have been four, differs from all others in that the envelope is rigid. Sixteen separate gas bags are contained in an aluminum alloy framework having 16 sides, covered with a cotton and rubber fabric. The pressure of the air is taken up by this framework instead of by the gas bags. The gas bags are not entirely filled, thus leaving room for expansion.

79 The rigid frame is 446 ft. long, 42½ ft. in diameter, and has ogival-shaped ends. It is braced about every 45 ft. by a number of rods crossing near the center, giving a cross section resembling a bicycle wheel. Vertical braces are placed at intervals the entire

length of the frame. The 16 gas bags are completely separated from each other by partitions of sheet aluminum. Under the framework is a triangular truss running nearly the entire length, the sides of the triangle being about 8 ft. The total volume of the gas bags is 460 000 cu. ft. which gives a gross lift of about 32 000 lb.

80 *Suspension.* The two cars are rigidly attached directly to the frame of the envelope, and a very short distance below it.

81 *Cars.* The two cars are built like boats. They are about 20 ft. long, 6 ft. wide, $3\frac{1}{3}$ ft. high; are placed about 100 ft. from each end and are made of the same aluminum alloy. To land the airship, it is lowered until the cars float on the water, when it can be towed like a ship. A third car is built into the keel directly under the center of the framework, and is for passengers only.

82 *Motors.* The power is furnished by two 110 h.p. Daimler-Mercedes motors, one placed on each car. Each weighs about 550 lb.; sufficient fuel for a 60 hours run can be carried.

83 *Propellers.* A pair of three-bladed metal propellers about 15 ft. in diameter is placed opposite each car, firmly attached to the frame of the envelope at the height of the center of resistance where they are most efficient.

84 *Stability.* In addition to the long V-shaped keel under the rigid frame, on each side at the rear of the frame are two nearly horizontal planes, while above and below the rear end are vertical fins.

85 *Steering.* A large vertical rudder is attached at the extreme end of the rigid frame, and an additional one is placed between each set of horizontal planes on the sides. For vertical steering, there are four sets of movable horizontal planes placed near the ends of the rigid frame, about the height of the propellers. Each set consists of four horizontal planes placed one above the other and connected with rods, so that they work on the principle of a shutter. These horizontal rudders serve another very important purpose, due to the reaction of the air. When these planes are set at an angle of 15 deg. and the airship is making a speed of 35 miles per hour, an upward pressure of over 1700 lb. is exerted, and consequently all the gas in one compartment could escape and yet by the manipulation of these planes, the airship could return safely to its starting point.

86 Its best performances were two long trips made during the past summer. The first, July 4, lasted exactly twelve hours, during which time it covered a distance of 235 miles, crossing the mountains to Lucerne and Zurich, and returning to the balloon house at Fried-

richshafen on Lake Constance. The average speed on this trip was 32 miles per hour. On August 4 this airship attempted a 24-hour flight, which was one of the requirements made for its acceptance by the Government. It left Friedrichshafen in the morning with the intention of following the Rhine as far as Mainz, and then returning to its starting point straight across the country. A stop of 4 hours and 30 minutes was made in the afternoon of the first day on the Rhine, to repair the engine. On the return, a second stop was found necessary near Stuttgart, due to difficulties with the motors and the loss of gas. While anchored to the ground a storm came up, and broke loose the anchorages; and as the balloon rose in the air it exploded and took fire, due to causes which have never been actually determined and published, and fell to the ground, resulting in its complete destruction. On this journey, which lasted in all 31 hours and 15 minutes, the airship was in the air 20 hours and 45 minutes, and covered a total distance of 378 miles.

87 The patriotism of the German nation was aroused. Subscriptions were immediately opened and in a short space of time $1 000 000 had been raised. A Zeppelin Society was formed to direct the expenditure of this fund. $85 000 has been expended for land near Friedrichshafen; shops are being constructed and it has been announced that within one year, the construction of 8 airships of the Zeppelin type will be completed. Recently the Crown Prince of Germany made a trip in the *Zeppelin No. 3*, which had been called back into service, and within a very few days the Emperor of Germany visited Friedrichshafen for the purpose of seeing the airship in flight. He decorated Count Zeppelin with the Order of the Black Eagle. German patriotism and enthusiasm has gone further, and the "German Association for an Aërial Fleet" has been organized in sections throughout the country. It announces its intention of building fifty garages (hangars) for housing airships.

THE UNITED STATES

SIGNAL CORPS DIRIGIBLE NO. 1

88 Due to lack of funds, the United States Government has not been able to undertake the construction of an airship sufficiently large and powerful to compete with those of European nations. However, specifications were sent out last January for an airship not over 120 ft. long and capable of making 20 miles per hour. Contract

was awarded to Capt. Thomas S. Baldwin, who delivered an airship last August to the Signal Corps, the description of which follows:

89 *Gas Bag.* The gas bag is spindle shaped, 96 ft. long, maximum diameter 19 ft. 6 in. with a volume of 20 000 cu. ft. A ballonet for air is provided inside the gas bag, and has a volume of 2800 cu. ft. The material for the gas bag is made of two layers of Japanese silk, with a layer of vulcanized rubber between.

90 *Car.* The car is made of spruce, and is 66 ft. long, 2½ ft. wide, and 2½ ft. high.

91 *Motor.* The motor is a 20 h.p. water-cooled Curtiss make.

92 *Propeller.* The propeller is at the front end of the car, and is connected to the engine by a steel shaft. It is built up of spruce, has a diameter of 10 ft. 8 in. with a pitch of 11 ft., and turns at the rate of 450 r.p.m. A fixed vertical surface is provided at the rear end of the car to minimize veering, and a horizontal surface attached to the vertical rudder at the rear tends to minimize pitching. A double horizontal surface controlled by a lever and attached to the car in front of the engine, serves to control the vertical motion and also to minimize pitching.

93 The position of the car very near to the gas bag, is one of the features of the Government dirigible. This reduces the length and consequently the resistance of the suspension, and places the propeller thrust near the center of resistance.

94 The total lifting power of this airship is 1350 lb. of which 500 lb. are available for passengers, ballast, fuel, etc. At its official trials a speed of 19.61 miles per hour was attained over a measured course, and during an endurance run lasting 2 hours, 70 per cent of the maximum speed was maintained.

95 *Dirigible No.* 1, as this airship has been named, has already served a very important purporse in initiating officers of the Signal Corps in the construction and operation of a dirigible balloon. With the experience now acquired, the United States Government is in a position to proceed with the construction and operation of an airship worthy of comparison with any now in existence, but any efforts in this direction must await the action of Congress in providing the necessary funds.

BALLOON PLANT AT FORT OMAHA, NEBRASKA

96 In anticipation of taking up the subject of aëronautics on a scale commensurate with its importance, a complete plant has been

constructed at the Signal Corps post at Fort Omaha, Nebraska. This plant comprises a steel balloon house 200 ft. long, 84 ft. wide, and 75 ft. high; that is, large enough to house a dirigible balloon of the size of the new French Military Airship *Republique.* For furnishing hydrogen gas, an electrolytic plant has been installed capable of furnishing 3000 cu. ft. of gas per hour. A gasometer of 50 000 cu. ft. capacity has been provided to store a sufficient supply of gas for emergencies.

97 In connection with the hydrogen plant, is a compressor for charging under pressure the steel tubes in which the gas is transported. A hydraulic pump for testing steel tubes at high pressure is a part of this equipment. A steel wireless telegraph tower 200 ft. high has been completed, and probably will be used in connection with wireless experiments with dirigible balloons.

Some General Considerations Which Govern the Design of a Dirigible Balloon

BUOYANCY AND SHAPE

98 Although many aërodynamic data are needed for the proper design of a dirigible airship, yet the experience already available in the construction and performance of such ships built on different plans is sufficient to enable the engineer to proceed with the design of a dirigible balloon to accomplish definite results along fairly accurate lines. In the case of this class of lighter-than-air ships the following general equation obtains:

$$W - w = V\left(\sigma - \frac{\sigma}{n}\right) \tag{1}$$

where

W = weight of balloon, envelope, car, and aëronauts
V = volume of balloon
σ = density of the air
n = density of air as compared with gas
w = weight of air displaced by car and aëronauts and envelope of balloon.

99 If we call the weight of the gas in the balloon M, then we can write this equation in the following manner:

$$W + M = w + nM$$

from which we find that

$$M = \frac{W - w}{n - 1} \tag{2}$$

and

$$V = \left(\frac{W - w}{\sigma}\right)\left(\frac{n}{n - 1}\right) \tag{3}$$

thus obtaining the volume of gas required. If the volume of the gas-bag, car, aëronauts, etc. $= v$, then $w = v\sigma$; so that (3) may be written

$$V = \left(\frac{W - v\sigma}{\sigma}\right)\left(\frac{n}{n - 1}\right) \tag{4}$$

100 Thus far, certainly, no dirigible balloon has ever been developed which has attained an independent speed greater than 40 miles per hour. It will readily be admitted that an airship so designed as to reach a speed of 50 or 60 miles per hour would be regarded as a most decided step forward in the art, since this difference of velocity is just the increment needed to place such craft on a practical basis capable of maneuvering in the air in all ordinary weather. This advancement, although requiring much consideration, would fully compensate in practical results.

101 The first point to be decided upon in the design of an airship is the method of maintaining the shape of the gas-bag against the pressure encountered at the maximum velocity to be attained. There are two schools of design in this respect, each having its adherents. One maintains the shape of the gas-bag by a rigid interior frame, and the other by means of the internal pressure of the gas itself.

102 Upon the selection of the type depends to a large extent the particular shape of the envelope. If the envelope is to maintain its shape by interior pressure of gas, evidently it must be so designed that the maximum pressure of the air developed at the speed contemplated shall not be sufficient to cause deformation of any part of the envelope. This can be effected only by making the uniform internal pressure at least equal to the maximum external pressure. Since the maximum external pressure occurs over the prow of the air-ship, this evidently is the particular part which must receive most careful attention with this system.

103 The desirable shape of head would evidently be one where the distribution of external pressure due to air resistance at the

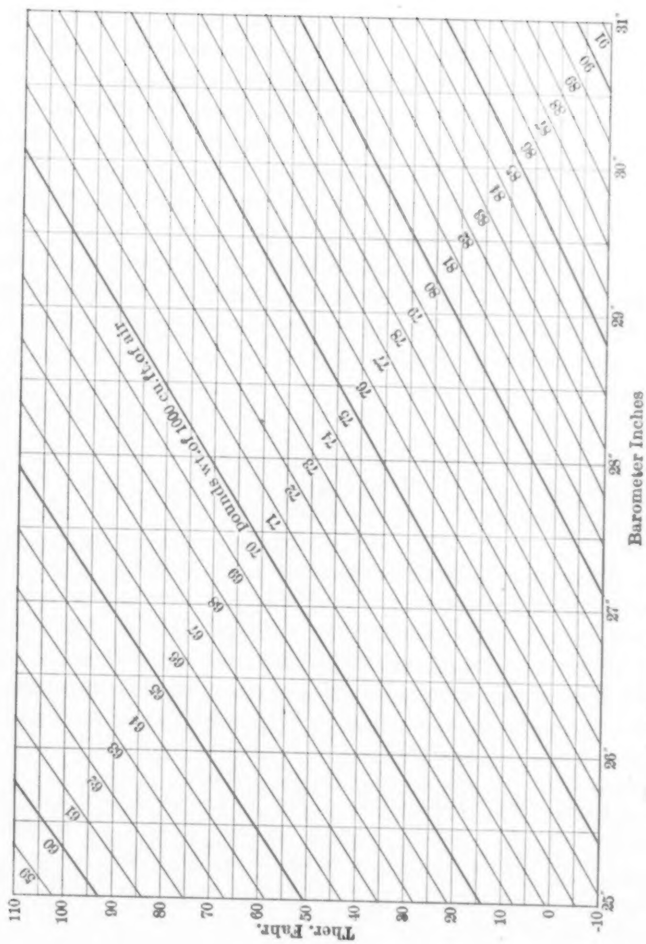

FIG. A DIAGRAM FOR FINDING THE ASCENSIONAL FORCE OF GASES

velocity used is uniform. In addition to preventing deformation of the gas-bag, a prime requisite also is that the shape shall be such that the total resistance, comprising head resistance and skin-friction, shall be a minimum for a given displacement and velocity.

104 This immediately forces the question of the law of resistance of the air. On this subject there are numerous aërodynamic data for low velocities, and also for very high velocities, but such data are incomplete for the range of velocities here considered.

105 In fact, the law of resistance of the air for surfaces of revolution, as experimentally determined, is known to vary not with any constant power of the velocity, but by a range of exponents from the first to the cube, if not higher. For example, in the enormous velocities attained by modern artillery, where bodies weighing a ton or more, are hurled through the air at 2000 ft. per second, it is known that the physical phenomena become entirely different in nature from those found when dealing with moderate velocities such as are met in transportation devices.

RESISTANCE OF THE AIR TO THE MOTION OF A PROJECTILE

106 In the expression for the retardation of oblong projectiles the velocity enters with an exponent, n, whose accepted values are as follows:

		Ft. per second	Miles per hour
$n = 1.55$	for velocities greater than	2600 =	1773
$n = 1.7$	for velocities between....	2600 and 1800 =	1773 and 1227
$n = 2.$	for velocities between....	1800 and 1370 =	1227 and 934
$n = 3.$	for velocities between....	1370 and 1230 =	934 and 836
$n = 5.$	for velocities between....	1230 and 970 =	836 and 639
$n = 3.$	for velocities between....	970 and 790 =	639 and 592
$n = 2.$	for velocities less than ...	790 =	592

107 *14-in. and 16-in. Guns.* The 14-in. guns fire a projectile weighing 1660 lb. Service muzzle velocity 2150 f. s., which gives with an elevation of 15 deg. a range of 15 000 yds.

108 The 16-in. guns fire a projectile weighing 2400 lb. The service muzzle velocity is 2150 f. s., or 1465 miles per hour, which gives, for an elevation of 15 deg., a range of 15 558 yd., or nearly 9 miles.

ANALOGY TO THE AIRSHIP

109 Great guns are now constructed which throw masses of steel weighing as high as 2400 lb. to maximum distances approximating

15 to 20 miles, and with such high momentum that ordinary winds have little effect, as shown by the remarkable target practice of the army and navy. The shapes of these heavier-than-air flying machines are figures of revolution, and the longitudinal and lateral stability are maintained by imparting to the projectile a rotary motion about its longer axis by means of the rifling inside the bore of the gun. Such machines are 5000 or 6000 times heavier than air and travel at speeds far beyond any other engine constructed by man. No peripheral speeds attained with any machinery approach these velocities.

110 It is noted that these projectile air-machines have a mass two and a half times that of the Wright aëroplane, and attain a velocity through the air thirty-six times as great.

111 It thus appears that the resistance of the air to the motion of bodies through it is in reality a complicated function of the velocity, and the best that can be said is that this velocity varies as a constant power only within certain limited ranges. In the velocities considered for airships, it is approximate to regard the resistance as varying as the square.

112 As the velocity increases the form of the head becomes more and more important, and moderate velocities lead to a shape approximating torpedo form, which is well known. In very high speed projectiles the shape of the rear is not so important, since the velocity is so much greater than the velocity of sound in air, that a partial vacuum is formed behind the projectile which cannot well be obviated.

113 If the rigid system be employed where an internal frame prevents deformation of the envelope, the stresses due to external pressure are taken up by the framework itself, and the gas required for flotation is usually contained in several separate receptacles or ballonets similar to compartments employed in ships. In this system, therefore, we are concerned only in securing such a shape of the rigid frame as will fulfill the condition of minimum total resistance for a given displacement and velocity.

114 Once the shape of the bag is determined from the considerations already enumerated, the dimensions become immediately fixed when the tonnage is assumed, or conversely, if any linear dimension is assigned the tonnage is thereby determined.

115 In addition to the two general systems above considered, there are various types involving some of the principles of each, which are classed in general as semi-rigid systems. Such systems usually comprise a rigid frame, to which is attached the gas-bag above, and the load below.

AËRODYNAMIC ADJUSTMENTS

116 The next step is one of structural design along strictly engineering lines. The aërodynamic features of airship construction may be considered under the heads: (*a*) static balance; (*b*) dynamic balance; (*c*) stability; (*d*) natural period and oscillation.

117 *Static Balance.* The dimensions of the gas-bag being determined, the lift of each transverse segment thereof is immediately known, and the design of the frame may proceed by approximate trial and correction as in other structural work. The weight of each segment of the envelope itself is readily computed, which added to the corresponding segment of the frame, gives the total weight of each segment, and this total subtracted from the lift of each segment gives the net lift for that complete segment. From the magnitude and position of these net forces the position of the resultant lift is known, and this determines the vertical line through the center of gravity. Such procedure evidently insures static balance of the machine as a whole, and an approximate distribution of the load.

118 *Dynamic Balance.* The dynamic balance must also be carefully considered; and here a difficulty has been experienced on account of the inability to place the resultant thrust coincident with the line of resistance of the ship as a whole. Heretofore, it has been customary to balance the thrust-resistance couple by means of suitable horizontal rudders or planes, so situated and at such angles, that the resultant moment of the system should be zero at uniform speeds of travel, though not necessarily zero for accelerated motion.

119 If, however, the line of thrust be made coincident with the line of resistance, the disturbing moment in question will be eliminated at uniform speeds. If, furthermore, the center of mass be located on the line of thrust and sufficiently forward to form a righting couple with the resistance when the wind suddenly veers, the evil effects of a disturbing moment will be obviated for variable as well as for constant speeds. The ship is then dynamically balanced.

120 This, of course, requires that the form of hull be such that a quartering wind shall exert a force passing to the rear of the center of mass. To illustrate, a good example of dynamic balance is found in a submarine torpedo, or a fish.

121 *Stability.* The foregoing adjustments still allow the center of mass to be placed below the center of buoyancy. This is a provision that is important in aëronautics as well as in marine architecture, indeed it is the only practical provision for keeping an even

keel and preventing heeling when the ship is at rest, or simply drifting with the wind. If the center of gravity be well below the center of buoyancy, the vessel is proportionately stable, but, of course, the stability is pendular, and may admit of considerable rolling and pitching due to shifting loads, sudden gusts of wind, etc., unless special devices be used to dampen or prevent these effects.

122 *Natural Period and Oscillation.* It may happen also that the equilibrium of the ship is disturbed by periodic forces whose periods are simply related to the natural period of the ship itself. In this case the oscillations will be cumulative and may become very large. Such effects are well known to marine engineers, and may be treated as in ordinary ship design.

AVIATION

123 This division comprises all those forms of heavier-than-air flying machines which depend for their support upon the dynamic reaction of the atmosphere. There are several subdivisions of this class dependent upon the particular principle of operation. Among these may be mentioned the aëroplane, orthopter, helicopter, etc. The only one of these that has been sufficiently developed at present to carry a man in practical flight is the aëroplane. There have been a large number of types of aëroplanes tested with more or less success and of these the following are selected for illustration.

Representative Aëroplanes of Various Types

THE WRIGHT BROTHERS' AËROPLANE

124 The general conditions under which the Wright machine was built for the Government were, that it should develop a speed of at least 36 miles per hour, and in its trial flights remain continuously in the air for at least 1 hour. It was designed to carry two persons having a combined weight of 350 lb. and also sufficient fuel for a flight of 125 miles. The trials at Fort Myer, Virginia, in September 1908, indicated that the machine was able to fulfill the requirements of the government specifications.

125 The aëroplane has two superposed main surfaces 6 ft. apart with a spread of 40 ft. and a distance of 6½ ft. from front to rear. The area of this double supporting surface is about 500 sq. ft. The surfaces are so constructed that their extremities may be warped at the will of the operator.

126 A horizontal rudder of two superposed plane surfaces about

15 ft. long and 3 ft. wide is placed in front of the main surfaces. Behind the main planes is a vertical rudder formed of two surfaces trussed together about 5½ ft. long and 1 ft. wide. The auxiliary surfaces, and the mechanism controlling the warping of the main surfaces, are operated by three levers.

127 The motor, which was designed by the Wright brothers, has four cylinders and is water cooled. It develops about 25 h.p. at 1400 r.p.m. There are two wooden propellers 8½ ft. in diameter which are designed to run at about 400 r.p.m. The machine is supported on two runners, and weighs about 800 lb. A monorail is used in starting.

128 The Wright machine has attained an estimated maximum speed of about 40 miles per hour. On September 12, a few days before the accident which wrecked the machine, a record flight of 1 hour, 14 minutes, 20 seconds was made at Fort Myer, Virginia. Since that date Wilbur Wright, at Le Mans, France, has made better records; on one occasion remaining in the air for more than an hour and a half with a passenger.

129 A reference to the attached illustrations of this machine will show its details, its method of starting, and its appearance in flight.

THE HERRING AËROPLANE

130 The Signal Corps of the Army has contracted with A. M. Herring, of New York, to furnish an aëroplane under the conditions enumerated in the specification already referred to and shown in the appendix to this paper. Mr. Herring made technical delivery of his machine at the aëronautical testing ground at Fort Myer, Virginia, on October 13.

131 In compliance with the request of Mr. Herring the details of this machine will not be made public at present, but the official tests required under the contract will be conducted in public as has been the case with other aëronautical devices. Opportunity will be afforded any one to observe the machine in operation.

132 This machine embodies new features for automatic control and contains an engine of remarkable lightness per horse-power.

THE FARMAN AËROPLANE

133 The Farman flying machine has two superposed aëro-surfaces 4 ft. 11 in. apart with a spread of 42 ft. 9 in. and 6 ft. 7 in. from front to rear. The total sustaining surface is about 560 sq. ft.

134 A box tail 6 ft. 7 in. wide and 9 ft. 10 in. long in rear of the main surfaces is used to balance the machine. The vertical sides of the tail are pivoted along the front edges, and serve as a vertical rudder for steering in a horizontal plane. There are two parallel, vertical partitions near the middle of the main supporting surfaces, and one vertical partition in the middle of the box tail. A horizontal rudder in front of the machine is used to elevate or depress it in flight.

135 The motor is an eight cylinder Antoinette of 50 h.p. weighing 176 lb., and developing about 38 h.p. at 1050 r.p.m.

136 The propeller is a built-up steel frame covered with aluminum sheeting, 7½ ft. in diameter, with a pitch of 4 ft. 7 in. It is mounted directly on the motor shaft immediately in rear of the middle of the main surfaces.

137 The framework is of wood, covered with canvas. A chassis steel tubing carries two pneumatic-tired bicycle wheels. Two smaller wheels are placed under the tail. The total weight of the machine is 1166 lb. The main surfaces support a little over two pounds per square foot. The machine has shown a speed of about 28 miles per hour and no starting apparatus is used.

138 On January 13, 1908, Farman won the *Grand Prix* of the Aëro Club of France in a flight of 1 minute and 28 seconds, in which he covered more than a kilometer. It is reported that on October 30, 1908, a flight of 20 miles, from Mourmelon to Rheims, was made with this machine.

THE BLERIOT AËROPLANE

139 Following Farman's first flight from town to town, M. Bleriot with his monoplane aëroplane made a flight from Toury to the neighborhood of Artenay and back, a total distance of about 28 kilometers. He landed twice during these flights and covered 14 kilometers of his journey in about 10 minutes, or attained a speed of 52 miles an hour.

THE JUNE BUG

140 The *June Bug* was designed by the Aërial Experiment Association, of which Alexander Graham Bell is president. It has two main superposed aërosurfaces with a spread of 42 ft. 6 in., including wing tips, with a total supporting surface of 370 sq. ft.

141 The tail is of the box type. The vertical rudder above the rear edge of the tail is 30 in. square. The horizontal rudder in front of the main surfaces is 30 in. wide by 8 ft. long. There are four

triangular wing tips pivoted along their front edges for maintaining transverse equilibrium. The vertical rudder is operated by a steering wheel, and the movable tips by cords attached to the body of the aviator.

142 The motor is a 25 h.p., 8 cylinder, air cooled, Curtiss. The single wooden propeller immediately behind the main surfaces is 6 ft. 2 in. in diameter and mounted directly on the motor shaft. It has a pitch angle of about 17 deg. and is designed to run at about 1200 r.p.m.

143 The total weight of the machine, with aviator, is 650 lb. It has a load of about $1\frac{3}{4}$ lb. per sq. ft. of supporting surface. Two pneumatic-tired bicycle wheels are attached to the lower part of the frame.

144 With this machine, Mr. G. H. Curtiss, on July 4, 1908, who the Scientific American trophy by covering the distance of over a mile in 1 minute and $42\frac{2}{5}$ seconds at a speed of about 39 miles per hour.

Some General Considerations Which Govern the Design of an Aëroplane

145 The design of an aëroplane may be considered under the heads of support, resistance and propulsion, stability and control.

146 *Support.* In this class of flying machines, since the buoyancy is practically insignificant, support must be obtained from the dynamic reaction of the atmosphere itself. In its simplest form, an aëroplane may be considered as a single plane surface moving through the air. The law of pressure on such a surface has been determined and may be expressed as follows:

$$P = 2\,k\sigma\,AV^2 \sin\alpha \qquad (1)$$

in which P is the normal pressure upon the plane, k is a constant of figure, σ the density of the air, A the area of the plane, V the relative velocity of translation of the plane through the air, and α the angle of flight.

147 This is the form taken by Duchemin's formula for small angles of flight such as are usually employed in practice. The equation shows that the upward pressure on the plane varies directly with the area of the plane, with the sine of the angle of flight, with the density of the air, and also with the square of the velocity of translation.

148 It is evident that the total upward pressure developed must be at least equal to the weight of the plane and its load, in order to support the system. If P is greater than the weight the machine will ascend; if less, it will descend.

149 The constant k depends only upon the shape and aspect of the plane, and should be determined by experiment. For example, with a plane one foot square $k\sigma = 0.00167$, as determined by Langley, when P is expressed in pounds per square foot, and V in feet per second.

Equation (1) may be written

$$A\,V^2 = \frac{P}{2\,k\sigma \sin\alpha}$$

If P and α are kept constant then the equation has the form

$$A\,V^2 = \text{constant} \tag{2}$$

PRINCIPLE OF REEFING IN AVIATION

150 An interpretation of (2) reveals interesting relations. The supporting area varied inversely as the square of the velocity. For example, in the Wright aëroplane, the supporting area at 40 miles per hour is 500 sq. ft., while if the speed is increased to 60 miles per hour this area need be only $\frac{500}{1.5^2} = 222$ sq. ft., or less than one-half of its present size. At 80 miles per hour the area would be reduced to 125 sq. ft., and at 100 miles per hour only 80 sq. ft. of supporting area is required. These relations are conveniently exhibited graphically.

151 It thus appears that if the angle of flight be kept constant in the Wright aëroplane, while the speed is increased to one hundred miles per hour, we may picture a machine which has a total supporting area of 80 sq. ft., or double surfaces each measuring about 2½ by 16 ft., or 4 by 10 ft. if preferred. Furthermore, the discarded mass of the 420 sq. ft. of the original supporting surface may be added to the weight of the motor and propellers in the design of a reduced aëroplane, since in this discussion the total mass is assumed constant at 1000 lb.

152 In the case of a bird's flight, its wing surface is "reefed" as its velocity is increased, which instinctive action serves to reduce its head resistance and skin-frictional area, and the consequent power required for a particular speed.

153 *Determination of k for Arched Surfaces.* Since arched surfaces are now commonly used in aëroplane construction, and as the above equation (1) applies to plane surfaces only, it is important to determine experimentally the value of the coefficient of figure k, for each type of arched surface employed, especially as k is shown in some cases to vary with the angle of flight α; i.e. the inclination of the chord of the surface to the line of translation.

154 Assuming α constant, however, we may compare the lift of any particular arched surface with a plane surface of the same projected plan and angle of flight.

155 To illustrate, in the case of the Wright aëroplane, let us assume

$$
\begin{aligned}
P &= 1000 \text{ lb.} = \text{total weight} = W. \\
A &= 500 \text{ sq. ft.} \\
V &= 40 \text{ miles per hour} = 60 \text{ ft. per second.} \\
\alpha &= 7 \text{ deg. approximately.}
\end{aligned}
$$

Whence
$$
\begin{aligned}
k\sigma &= \frac{P}{2\,A\,V^2 \sin \alpha} = \frac{1000}{2 \times 500 \times 60^2 \times \frac{1}{8}} \\
&= 0.0022 \ (V = \text{ft. sec.}) \\
&= 0.005 \ \ (V = \text{mi. hr.})
\end{aligned}
$$

156 Comparing this value of $k\,\sigma$ with Langley's value 0.004 for a plane surface, V being in miles per hour, we see that the lift for the arched surface is 25 per cent greater than for a plane surface of the same projected plan. That is to say, this arched surface is dynamically equivalent to a plane surface of 25 per cent greater area than the projected plan. Such a plane surface may be defined as the "equivalent plane."

157 *Resistance and Propulsion.* The resistance of the air to the motion of an aëroplane is composed of two parts: (*a*), the resistance due to the framing and load; (*b*), the necessary resistance of the sustaining surfaces that is, the drift, or horizontal component of pressure and the unavoidable skin-friction. Disregarding the frame, and considering the aëroplane as a simple plane surface, we may express the resistance by the equation

$$
R = W \tan \alpha + 2\,f\,A \tag{3}
$$

in which R is the total resistance, W the gross weight sustained, α the angle of flight, f the friction per square unit of area of the plane,

A the area of the plane. The first term of the second member gives the drift, the second term the skin-friction. The power required to propel the aëroplane is

$$H = R\ V$$

in which H is the power, V the velocity.

158 Now W varies as the second power of the velocity, as shown by equation (1), and f varies as the power 1.85, as will be shown later. Hence we conclude that the total resistance R of the air to the aëroplane varies approximately as the square of its speed, and the propulsive power practically as the cube of speed.

159 *Most Advantageous Speed and Angle of Flight.* Again, regarding W and A as constant, we may, by equation (1), compute α for various values of V, and find f for those velocities from the skin-friction table to be given presently. Thus α, R, and H may be found for various velocities of flight, and their magnitudes compared. In this way the values in Table 1 were computed for a soaring plane 1 ft. square weighing 1 lb., assuming $k\ \sigma = 0.004$, which is approximately Langley's value when V is in miles per hour.

TABLE 1 POWER REQUIRED TO TOW A PLANE ONE FOOT SQUARE, WEIGHING ONE POUND, HORIZONTALLY THROUGH THE AIR AT VARIOUS SPEEDS AND ANGLES OF FLIGHT

COMPUTED

Velocity	Angle of flight	Computed Resistance			Tow-line power	Lift per tow-line h.p.
		Drift	Friction	Total		
Mi. hr.	*Deg.*	*Lb.*	*Lb.*	*Lb.*	*Ft. lb. sec.*	*Lb.*
30	8.25	0.145	0.0170	0.162	7.13	77.1
35	5.94	0.104	0.0226	0.1266	6.51	84.3
40	4.52	0.790	0.0289	0.1079	6.32	86.7
45	3.55	0.0621	0.0360	0.0981	6.39	86.1
50	2.88	0.0500	0.0439	0.0939	6.89	80.2
60	2.03	0.0354	0.0614	0.0962	8.50	64.7
70	1.47	0.0257	0.0814	0.1071	11.00	50.0
80	1.12	0.0195	0.1045	0.1240	14.56	35.8
90	0.88	0.0154	0.1300	0.1454	19.17	28.7
100	0.71	0.0124	0.1584	0.1708	25.00	22.0

160 Column two, giving values of α for various speeds, is computed from equation (1). Thus at 30 miles per hour,

$$\sin \alpha = \frac{W}{2\ k\ \sigma\ AV^2} = \frac{1}{2 \times .004 \times 1 \times 30^2}$$

whence $\alpha = 8.25$ deg.

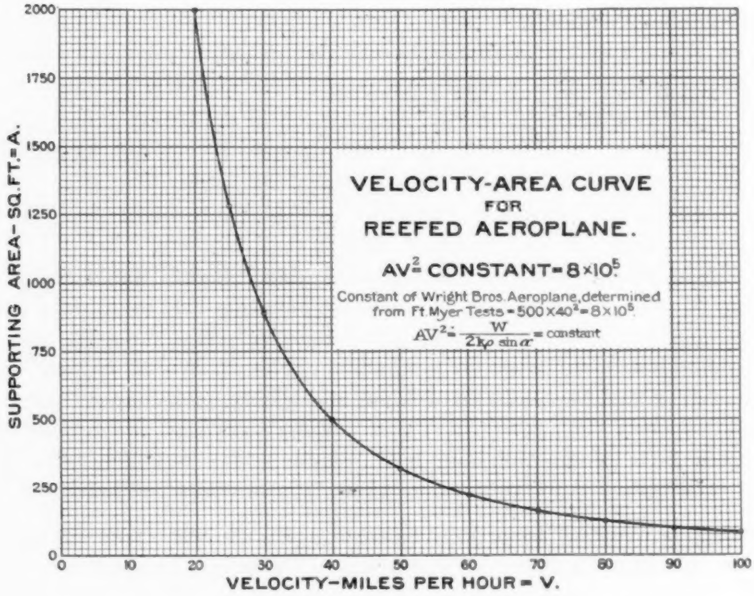
VELOCITY-AREA CURVE
FOR
REEFED AEROPLANE.
AV² = CONSTANT = 8×10⁵
Constant of Wright Bros. Aeroplane, determined from Ft. Myer Tests = 500×40² = 8×10⁵
SUPPORTING AREA - SQ. FT. = A.
VELOCITY-MILES PER HOUR = V.
0
250
500
750
1000
1250
1500
1750
2000
10
20
30
40
50
60
70
80
90
100

Fig. B

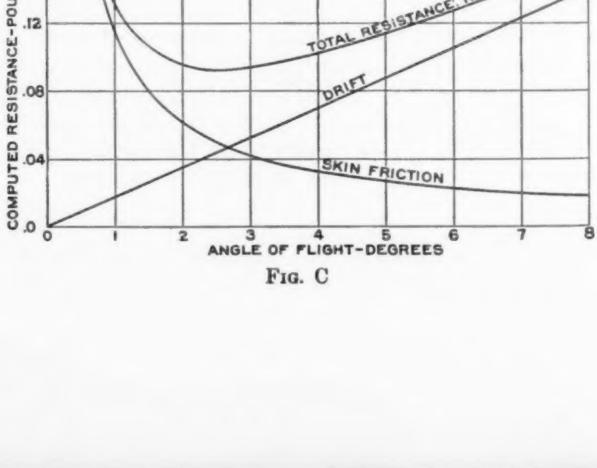
FOOT SQUARE PLANE WEIGHING ONE POUND
COMPUTED RESISTANCE-POUNDS
ANGLE OF FLIGHT-DEGREES
TOTAL RESISTANCE, R.
DRIFT
SKIN FRICTION
.0
.04
.08
.12
.16
0
1
2
3
4
5
6
7
8

Fig. C

161 Column three is computed from the term $W \tan \alpha$ in equation (3), thus

$$\text{Drift} = W \tan \alpha = 1 \times \tan 8.25 \text{ deg.} = 0.145.$$

162 Column four is computed from the term $2fA$, in equation (3), f being taken from the skin-friction table, to be given presently.

163 The table shows that if a thin plane 1 ft. square, weighing 1 lb. be towed through the air so as just to float horizontally at various velocities and angles of flight, the total resistance becomes a minimum at an angle of slightly less than 3 deg., and at a velocity of about 50 miles per hour; also that the skin-friction approximately equals the drift at this angle. The table also shows that the propulsive power for the given plane is a minimum at a speed of between 40 and 45 miles per hour, the angle of flight then being approximately 4.5 deg.

164 The last column of the table shows that the maximum weight carried per horse-power is less than 90 lb. This horse-load may be increased by changing the foot square plane to a rectangular plane and towing it long-side foremost; also by lightening the load, and letting the plane glide at a lower speed; but best of all, perhaps, by arching it like a vulture's wing and also towing it long-side foremost as is the prevailing practice with aëroplanes.

These relations are exhibited graphically in the diagrams, Fig. C, D and E.

STABILITY AND CONTROL

165 The question of stability is a serious one in aviation, especially as increased wind velocities are encountered. In machines of the aëroplane type there must be some means provided to secure fore and aft stability and also lateral stability.

166 A large number of plans have been proposed for the accomplishment of these ends, some based upon the skill of the aviator, others operated automatically, and still others employing a combination of both. At the present time no aëroplane has yet been publicly exhibited which is provided with automatic control. There is little difference of opinion as to the desirability of some form of automatic control.

167 The Wright aëroplane does not attempt to accomplish this, but depends entirely upon the skill of the aviator to secure both lateral and longitudinal equilibrium, but it is understood that a

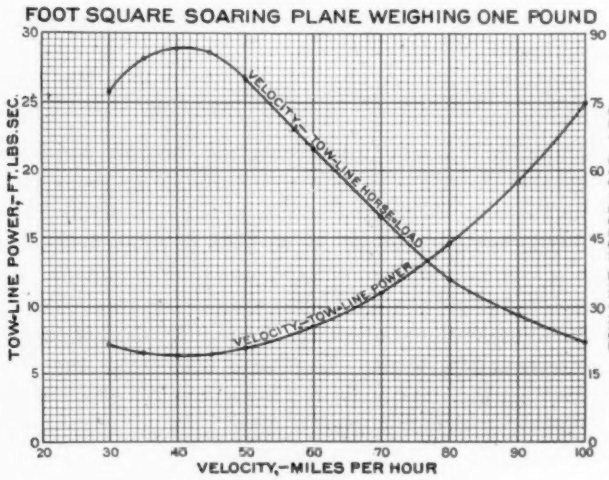
FOOT SQUARE SOARING PLANE WEIGHING ONE POUND
TOW-LINE POWER,-FT.LBS.SEC.
TOW-LINE HORSE-LOAD,-LBS.
VELOCITY,-TOW-LINE HORSE-LOAD
VELOCITY,-TOW-LINE POWER
VELOCITY,-MILES PER HOUR

FIG.

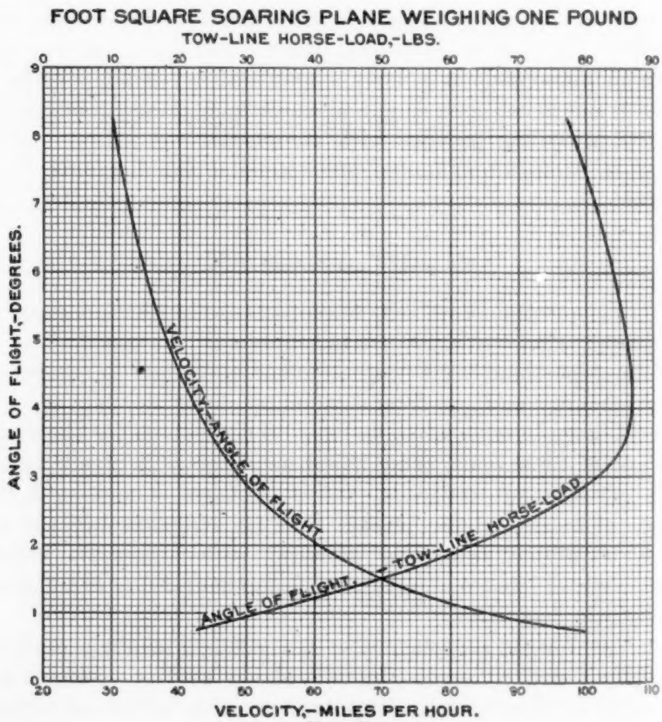
FOOT SQUARE SOARING PLANE WEIGHING ONE POUND
TOW-LINE HORSE-LOAD,-LBS.
ANGLE OF FLIGHT,-DEGREES.
VELOCITY,-ANGLE OF FLIGHT
ANGLE OF FLIGHT,-TOW-LINE HORSE-LOAD
VELOCITY,-MILES PER HOUR.

FIG. E

device for this purpose is one of the next to be brought forward by them. Much of the success of the Wright brothers has been due to their logical procedure in the development of the aëroplane, taking the essentials, step by step, rather than attempting everything at once, as is so often the practice with inexperienced inventors.

168 The aviator's task is much more difficult than that of the chauffeur. With the chauffeur, while it is true that it requires his constant attention to guide his machine, yet he is traveling on a roadway where he can have due warning, through sight, of the turns and irregularities of the course.

169 The fundamental difference between operating the aëroplane and the automobile, is that the former is traveling along an aërial highway which has manifold humps and ridges, eddies and gusts, and since the air is invisible he cannot see these irregularities and inequalities of his path, and consequently cannot provide for them until he has actually encountered them. He must feel the road since he cannot see it.

170 Some form of automatic control whereby the machine itself promptly corrects for the inequalities of its path is evidently very desirable. As stated above, a large number of plans for doing this have been proposed, many of them based on gyrostatic action, movable side planes, revolving surfaces, warped surfaces, etc. A solution of this problem may be considered as one of the next important steps forward in the development of the aëroplane.

HYDROMECHANIC RELATIONS

Some General Relations between Ships in Air and in Water

171 At the present moment, so many minds are engaged upon the general problem of aërial navigation that any method by which a broad forecast of the subject can be made is particularly desirable. Each branch of the subject has its advocates, each believing implicitly in the superiority of his method. On the one hand the adherents of the dirigible balloon have little confidence in the future of the aëroplane, while another class have no energy to devote to the dirigible balloon, and still others prefer to work on the pure helicopter principle. As a matter of fact, each of these types is probably of permanent importance, and each particularly adapted to certain needs.

172 Fortunately for the development of each type, the experiments made with one class are of value to the other classes, and these

in turn bear close analogy to the types of boats used in marine navigation. The dynamical properties of water and air are very much alike, and the equations of motion are similar for the two fluids, so that the data obtained from experiments in water, which are very extensive, may with slight modification be applied to computations for aërial navigation.

173 *Helmholz' Theorem.* Von Helmholz, the master physicist of Germany, who illuminated everything he touched, has fortunately considered this subject, in a paper written in 1873. The title of his paper is "On a theorem relative to movements that are geometrically similar in fluid bodies, together with an application to the problem of steering balloons."

174 In this paper Helmholz affirms that, although the differential equations of hydro-mechanics may be an exact expression of the laws controlling the motions of fluids, still it is only for relatively few and simple experimental cases that we can obtain integrals appropriate to the given conditions, particularly if the cases involve viscosity and surfaces of discontinuity.

175 Hence, in dealing practically with the motion of fluids, we must depend upon experiment almost entirely, often being able to predict very little from theory, and that usually with uncertainty. Without integrating, however, he applies the hydrodynamic equations to transfer the observations made on any one fluid with given models and speeds, over to a geometrically similar mass of another fluid involving other speeds, and models of different magnitudes. By this means he is able to compute the size, velocity, resistance, power, etc., of aërial craft, from given, or observed, values for marine craft.

176 He also deduces laws that must inevitably place a limit upon the possible size and velocity of aërial craft, without, however, indicating what that limit may be with artificial power. Applying this mode of reasoning to large birds he concludes by saying that, "It therefore appears probable that in the model of the great vulture, nature has already reached the limit that can be attained with the muscles as working organs, and under the most favorable conditions of subsistence, for the magnitude of a creature that shall raise itself by its wings and remain a long time in the air."

177 In comparing the behavior of models in water and air, he takes account of the density and viscosity of the media, as these were well known at the date of his writing, 1873; but he could not take account of the sliding, or skin-friction, because in his day neither

the magnitude of such friction for air, nor the law of its variation with velocity had been determined.

SKIN-FRICTION IN AIR

178 Even as late as Langley's experiments, skin-friction in air was regarded as a negligible quantity, but due to the work of Dr. Zahm, who was the first to make any really extensive and reliable experiments on skin-friction in air, we now can estimate the magnitude of this quantity. As a result of his research he has given in his paper on atmospheric friction the following equation:

$$f = 0.00000778\ l^{-0.07}\ v^{1.85} \ldots (v = \text{ft. sec.})$$
$$f = 0.0000158\ \ l^{-0.07}\ v^{1.85} \ldots (v = \text{mi. hr.})$$

in which f is the average skin-friction per square foot, and l the length of surface.

179 From this equation the accompanying table of resistances was computed, and is inserted here for the convenience of engineers:

TABLE 2 FRICTION PER SQUARE FOOT FOR VARIOUS SPEEDS AND LENGTHS OF SURFACE

Wind speed	Average Friction in Pounds per Square Foot					
	1 ft. plane	2 ft. plane	4 ft. plane	8 ft. plane	16 ft. plane	32 ft. plane
mi. hr.						
5	0.000303	0.000289	0.000275	0.000262	0.000250	0.000238
10	0.00112	0.00105	0.00101	0.000967	0.000922	0.000878
15	0.00237	0.00226	0.00215	0.00205	0.00195	0.00186
20	0.00402	0.00384	0.00365	0.00349	0.00332	0.00317
25	0.00606	0.00579	0.00551	0.00527	0.00501	0.00478
30	0.00850	0.00810	0.00772	0.00736	0.00701	0.00668
35	0.01130	0.0108	0.0103	0.0098	0.00932	0.00888
40	0.0145	0.0138	0.0132	0.0125	0.0125	0.0114
50	0.0219	0.0209	0.0199	0.0190	0.0181	0.0172
60	0.0307	0.0293	0.0279	0.0265	0.0253	0.0242
70	0.0407	0.0390	0.0370	0.0353	0.0337	0.0321
80	0.0522	0.0500	0.0474	0.0452	0.0431	0.0411
90	0.0650	0.0621	0.0590	0.0563	0.0536	0.0511
100	0.0792	0.0755	0.0719	0.0685	0.0652	0.0622

180 The numbers within the rules represent data coming within the range of observation. These observations show that "the frictional resistance is at least as great for air as water, in proportion to their densities. In other words, it amounts to a decided obstacle

in high speed transportation. In aëronautics it is one of the chief elements of resistance both to hull-shaped bodies and to aëro-surfaces gliding at small angles of flight."

181 *Relative Dynamic and Buoyant Support.* Peter Cooper-Hewitt has given careful study to the relative behavior of ships in air and in water. He has made a special study of hydroplanes, and has prepared graphic representations of his results which furnish a valuable forecast of the problem of flight.

182 Without knowing of Helmholz's theorem, Cooper-Hewitt has independently computed curves for ships and hydroplanes from actual data in water, and has employed these curves to solve analogous problems in air, using the relative densities of the two media, approximately 800 to 1, in order to determine the relative values of support by dynamic reaction and by displacement for various weights and speeds.

183 An analysis of these curves leadsto conclusions of importance, some of which are as follows:

184 The power consumed in propelling a displacement vessel at any constant speed, supported by air or water, is considered as being $\frac{2}{3}$ consumed by skin-resistance, or surface resistance, and $\frac{1}{3}$ consumed by head resistance. Such a vessel will be about ten diameters in length, or should be of such shape that the sum of the power consumed in surface friction and in head resistance will be a minimum (torpedo shape).

185 The power required to overcome friction due to forward movement will be about $\frac{1}{8}$ as much for a vessel in air as for a vessel of the same weight in water.

186 Leaving other things out of consideration, higher speeds can be obtained in craft of small tonnage by the dynamic reaction type than by the displacement type; for large tonnages the advantages of the displacement of type are manifest.

187 A dirigible balloon carrying the same weight, other things being equal, may be made to travel about twice as fast as a boat for the same power; or be made to travel at the same speed with the expenditure of about $\frac{1}{8}$ of the power.

188 As there are practically always currents in the air, reaching at times a velocity of many miles per hour, a dirigible balloon should be constructed with sufficient power to be able to travel at a speed of about 50 miles per hour, in order that it may be available under practical conditions of weather. In other words, it should have substantially as much power as would drive a boat, carrying the same

weight, 25 miles an hour, or should have the same ratio of power to size as the Lusitania.

189 *Motors.* It is the general opinion that any one of several types of internal combustion motors at present available is suitable for use with dirigible balloons. With this type, lightness need not be obtained at the sacrifice of efficiency. In the aëroplane, however, lightness per output is a prime consideration, and certainty and reliability of action is demanded, since if by chance the motor stops, the machine must immediately glide to the earth. A technical discussion of motors would of itself require an extended paper, and may well form the subject of a special communication.

190 *Propellers.* The fundamental principles of propellers are the same for air as for water. In both elements, the thrust is directly proportional to the mass of fluid set in motion per second. A great variety of types of propellers have been devised, but thus far only the screw-propeller has proved to be of practical value in air. The theory of the screw-propeller in air is substantially the same as for the deeply submerged screw-propeller in water, and therefore does not seem to call for treatment here. There is much need at present for accurate aërodynamic data on the behavior of screw-propellers in air, and it is hoped that engineers will soon secure such data, and present it in practical form for the use of those interested in airship design.

191 *Limitations.* Euclid's familiar "square-cube" theorem connecting the volumes and surfaces of similar figures, as is well known, operates in favor of increased size of dirigibles, and limits the possible size of heavier-than-air machines in single units and with concentrated load.

192 It appears, however, that both fundamental forms of aërial craft will likely be developed, and that the lighter-than-air type will be the burden-bearing machine of the future, whereas the heavier-than-air type will be limited to comparatively low tonnage, operating at relatively high velocity. The helicopter type of machine may be considered as the limit of the aëroplane, when by constantly increasing the speed, the area of the supporting surfaces is continuously reduced until it practically disappears. We may then picture a racing aëroplane propelled by great power, supported largely by the pressure against its body, and with its wings reduced to mere fins which serve to guide and steady its motion. In other words, starting with the aëroplane type; we have the dirigible balloon on the one hand as the tonnage increases, and the helicopter type on the other extreme as the speed increases. Apparently, therefore, no one of

these forms will be exclusively used, but each will have its place for the particular work required.

AËRIAL LOCOMOTION IN WARFARE

193 Whatever may be the influence of aërial navigation upon the art of war, the fact which must be considered at present is, that each of the principal military powers is displaying feverish activity in developing this auxiliary as an adjunct to the military establishment.

194 If each of the great Powers of the world would agree that aërial warfare should not be carried on, the subject would be of no great interest to this country as far as our military policy is concerned, but until such an agreement is made this country is forced to an immediate and serious consideration of this subject in order to be prepared for any eventuality.

195 The identical reasoning which has led to the adoption of a policy of providing for increasing our navy year by year to maintain our relative supremacy on the sea, is immediately applicable to the military control of the air. If the policy in respect to the navy is admitted, there is no escape from the deduction that we should proceed in the development of ships of the air on a scale commensurate with the position of the Nation.

196 The question as to whether or not the Powers will ultimately permit the use of aërial ships in war is not at present the practical one, because in case such use is authorized it will be too late adequately to equip ourselves after war has been declared.

ACTION OF THE HAGUE PEACE CONFERENCE

197 The following is the declaration signed by the delegates of the United States to the Second International Peace Conference held at The Hague, June 15 to October 19, 1907, prohibiting the discharge of projectiles and explosives from balloons, ratified March 10, 1908.

198 Declaration:

The Contracting Powers agree to prohibit, for a period extending to the close of the Third Peace Conference, the discharge of projectiles and explosives from balloons or by other new methods of a similar nature.

199 The delegates of the United States signed this declaration. The countries which did not sign the declaration forbidding the launching of projectiles and explosives from balloons were: Germany,

Austria-Hungary, China, Denmark, Ecuador, Spain, France, Great Britain, Guatemala, Italy, Japan, Mexico, Montenegro, Nicaragua, Paraguay, Roumania, Russia, Servia, Sweden, Switzerland, Turkey, Venezuela.

200 It appears that the United States is the only first-class Power who signed this agreement, and an analysis of the text of the agreement itself shows that no serious attempt was made to settle the question finally.

201 For instance, while the war balloon may not discharge projectiles or explosives from above, no reciprocal provision is ade preventing such war balloon from being fired upon from the earth below, yet the law of self-defense evidently obtains.

202 Furthermore, Naval Experts will tell you that they fear no enemy quite as much as a submarine mine, whose location is unknown and which gives no warning when it is approached. Our own experience shows that the Battleship Maine could be completely destroyed in time of peace without any one detecting the preparations for its accomplishment.

203 If, then, a nation can submerge a mine for the destruction of ships from underneath the water, why can it not drop an aërial mine upon a ship from above? And if it should be allowed to drop an aërial mine upon an enemy's fortified ship at sea, it certainly should be allowed to drop such an aërial mine upon a fortified place on land.

INFLUENCE ON THE MILITARY ART

204 The Military Art, up to the present time, has been practically conducted in a plane where the armies concerned have been limited in their movements in time and place by the physical character of the terrane. A large army, for instance, cannot move faster than about 12 miles a day by marching, and the use of railroads as applied to the Art of War was first recognized in the Franco-Prussian war. By their use, the mobilization of the great Prussian army, and its accurate assembling in the theater of operations within ten days, contributed an initial advantage not before possible.

205 The very essence of strategy is surprise, and there never were better opportunities than at present for a constructive general to achieve great victories. But these victories, to be really great, must be founded upon some new development or use of power not heretofore known in war. They must also tend to produce results with the minimum loss of human life. In other words, the sentiment of

the world demands that the military art shall always aim to capture, not destroy.

206 It may be said, that the consummation of military art is found in maneuvering the enemy into untenable situations, thereby forcing a decisive result with a minimum loss of life and treasure.

207 As to the technical use of dirigible balloons and aëroplanes in warfare we have nothing but theory at present to guide us. It would appear, however, in the case of dirigible balloons that two different classes of such ships should be developed.

208 First: A comparatively small dirigible type with a capacity of from 50 000 to 100 000 cubic feet, to be used principally for scouting purposes and to a limited extent for carrying explosives for demolitions or for incendiary purposes, such as destroying bridges and supply depots close to the mobile army or coast defense fortress. In reconnoitering dirigibles of this class, in order to be safe during day-time they will have to maneuver at an altitude of about a mile, but experiments show that telephotographic apparatus will operate from this height to give much detail.

209 At night, such dirigibles may descend to within a few hundred feet of the ground with safety and thus obtain much valuable information. Equipped with wireless telegraph or telephone apparatus, military data could be obtained and transmitted without undue risk. Due to the small carrying capacity of such sizes, the radius of action would probably be limited at present to about two hundred miles.

210 Second: This type of dirigible may be developed for burden-bearing purposes. It has been pointed out above that the larger the airship the greater the speed it may be given, and the greater its radius of action. There is no reason to doubt, that airships of a capacity of from 500 000 to 1 000 000 cubic feet may be ultimately developed to attain speeds of 50 to 75 miles per hour. With a capacity for such speed, the aërial craft becomes a powerful practical engine of war which may be used in all ordinary weather. By keeping high in the air in day-time, and descending at night, they may launch high explosives, producing great damage. Being able to pass over armies and proceed at great speeds, their objectives would not usually be the enemy's armies, but their efforts would be directed against his base of supplies; to destroy his dry-docks, arsenals, ammunition depots, principal railway centers, storehouses, and indeed the enemy's navy itself.

211 It is thought that there will be little difficulty in launching explosives with accuracy, provided good maps and plans are available. Due to the small cost of such ships as compared with naval vessels, the risk of loss would be readily taken.

212 The element of time has always been a controlling factor in warfare. It is often a military necessity to conduct a reconnoissance in force to develop the enemy's dispositions. This requires at times a detachment of several thousand men from the main army, for a considerable period of time. With efficient military airships, these results may be attained with a very few men in a small fraction of the time heretofore required.

213 *Delimitation of Frontiers.* The realization of aërial navigation for military purposes, brings forward new questions regarding the limitation of frontiers. As long as military operations are confined to the surface of the earth, it has been the custom to protect the geographical limits of a country by ample preparations in time of peace, such as a line of fortresses properly garrisoned. At the outbreak of war these boundaries represent real and definite limits to military operations. Excursions into the enemy's territory usually require the backing of a strong military force. Under the new conditions, however, these geographic boundaries no longer offer the same definite limits to military movements. With a third dimension added to the theater of operations, it will be possible to pass over this boundary on rapid raids for obtaining information, accomplishing demolitions, etc., returning to safe harbors in a minimum time. We may, therefore, regard the advent of military ships of the air, as, in a measure, obliterating present national frontiers in conducting military operations.

214 One of the military objectives in warfare, is usually the enemy's capital city, his ministers, and his chief executive. This objective has heretofore been protected by large armies of soldiers, who in themselves are not so important to the result. In order to attain the objective, it has been frequently necessary to subdue large numbers of soldiers needlessly.

215 With the advent of efficient ships of the air, however, small parties may pass over these protective armies on expeditions aimed at the seat of government itself, where reside the body of particular individuals most responsible, so that the ultimate result will be to deter a rash entrance into war for personal ends; since now for the first time responsible individuals of state may be in immediate and personal danger after the declaration of war, which heretofore has not been usually the case.

INTERIOR HARBORS

216 In the development of these larger types of dirigible balloons the main difficulty will be, in providing suitable harbors or places of safety, for replenishing supplies and for seeking shelter in times of stress. As long as the dirigible balloon remains in the air it may be regarded as tolerably safe, both in itself, and as a conveyance for observers. If its engines are disabled, it is at least a free balloon and may be operated as such.

217 When brought in contact with the ground, however, it is in considerable danger from high winds. The momentum of such an enormous airship is great, and the comparatively fragile structure of the craft makes it an easy prey to the pounding which it is likely to receive when landing. Just as marine ships must seek a sheltered harbor or put to the open sea in times of storm, so in case of ships of the air, it is much more necessary either to brave the storm in the open, or to seek some sheltered harbor on land.

218 Fortunately, in this case, certain suitable harbors for very large ships may be provided at small expense, by using narrow and deep valleys and ravines, surrounded by forests or other protection, or prepared railway cuts, etc., where the airship may descend and be reasonably safe from the winds above. These harbors should, of course, be known to the pilot, and carefully plotted on his maps beforehand. The compass bearings of each harbor from prominent points on land must be known and plotted, to assist as far as possible in navigating the airship in thick weather; and such harbors may be indicated to the pilot at night by vertical searchlight beams, or by suitable rockets, etc.

219 The aëroplane, as has been pointed out, is likely to prove a flying machine of comparatively low tonnage and high speed. It is not likely to become a burden-bearing ship, at least in single units, but will be extremely useful for reconnoitering purposes; for dispatching important orders and instructions at high speed; for reaching inaccessible points; or for carrying individuals of high rank and command to points where their personality is needed.

220 One of the bloodiest contests the world has ever seen, was the Japanese attack on "203 Meter Hill," yet the sole object of this great slaughter was to place two or three men at its summit to direct the fire of the Japanese siege guns upon the Russian fleet in the harbor at Port Arthur.

221 If the United States had possessed in 1898, a single dirigible

balloon, even of the size of the one now at Fort Myer, Virginia, which cost less than $10 000, the American Army and Navy would not have long remained in doubt of the presence of Cervera's fleet in Santiago Harbor.

222 The world is undoubtedly growing more humane year by year. We have arrived at a conception of the principle of an efficient army and navy, not to provoke war but to preserve peace, and it is believed, that, following this principle, the perfection of ships of the air for military purposes will materially contribute, on the whole, to make war less likely in the future than in the past.

APPENDIX NO. 1

SIGNAL CORPS SPECIFICATION, NO. 486

Advertisement and Specification for a Heavier-than-Air Flying Machine.

To the Public:

Sealed proposals, in duplicate, will be received at this office until 12 o'clock noon on February 1, 1908, on behalf of the Board of Ordnance and Fortification for furnishing the Signal Corps with a heavier-than-air flying machine. All proposals received will be turned over to the Board of Ordnance and Fortification at its first meeting after February 1 for its official action.

Persons wishing to submit proposals under this specification can obtain the necessary forms and envelopes by application to the Chief Signal Officer, United States Army, War Department, Washington, D. C. The United States reserves the right to reject any and all proposals.

Unless the bidders are also the manufacturers of the flying machine they must state the name and place of the maker.

Preliminary.—This specification covers the construction of a flying machine supported entirely by the dynamic reaction of the atmosphere and having no gas bag.

Acceptance.—The flying machine will be accepted only after a successful trial flight, during which it will comply with all requirements of this specification. No payments on account will be made until after the trial flight and acceptance.

Inspection.—The Government reserves the right to inspect any and all processes of manufacture.

GENERAL REQUIREMENTS.

The general dimensions of the flying machine will be determined by the manufacturer, subject to the following conditions:

1 Bidders must submit with their proposals the following:
- *a* Drawings to scale showing the general dimensions and shape of the flying machine which they propose to build under this specification.
- *b* Statement of the speed for which it is designed.
- *c* Statement of the total surface area of the supporting planes.
- *d* Statement of the total weight.
- *e* Description of the engine which will be used for motive power.
- *f* The material of which the frame, planes, and propellers will be constructed. Plans received will not be shown to other bidders.

2 It is desirable that the flying machine should be designed so that it may be quickly and easily assembled and taken apart and packed for transportation in army wagons. It should be capable of being assembled and put in operating condition in about one hour.

3 The flying machine must be designed to carry two persons having a combined weight of about 350 pounds, also sufficient fuel for a flight of 125 miles.

4 The flying machine should be designed to have a speed of at least forty miles per hour in still air, but bidders must submit quotations in their proposals for cost depending upon the speed attained during the trial flight, according to the following scale:

40 miles per hour, 100 per cent.
39 miles per hour, 90 per cent.
38 miles per hour, 80 per cent.
37 miles per hour, 70 per cent.
36 miles per hour, 60 per cent.
Less than 36 miles per hour rejected.
41 miles per hour, 110 per cent.
42 miles per hour, 120 per cent.
43 miles per hour, 130 per cent.
44 miles per hour, 140 per cent.

5 The speed accomplished during the trial flight will be determined by taking an average of the time over a measured course of more than five miles, against and with the wind. The time will be taken by a flying start, passing the starting point at full speed at both ends of the course. This test subject to such additional details as the Chief Signal Officer of the Army may prescribe at the time.

6 Before acceptance a trial endurance flight will be required of at least one hour during which time the flying machine must remain continuously in the air without landing. It shall return to the starting point and land without any damage that would prevent it immediately starting upon another flight. During this trial flight of one hour it must be steered in all directions without difficulty and at all times under perfect control and equilibrium.

7 Three trials will be allowed for speed as provided for in paragraphs 4 and 5. Three trials for endurance as provided for in paragraph 6, and both tests must be completed within a period of thirty days from the date of delivery. The expense of the tests to be borne by the manufacturer. The place of delivery to the Government and trial flights will be at Fort Myer, Virginia.

8 It should be so designed as to ascend in any country which may be encountered in field service. The starting device must be simple and transportable. It should also land in a field without requiring a specially prepared spot and without damaging its structure.

9 It should be provided with some device to permit of a safe descent in case of an accident to the propelling machinery.

10 It should be sufficiently simple in its construction and operation to permit an intelligent man to become proficient in its use within a reasonable length of time.

11 Bidders must furnish evidence that the Government of the United States has the lawful right to use all patented devices or appurtenances which may be a part of the flying machine, and that the manufacturers of the flying machine are authorized to convey the same to the Government. This refers to the unrestricted right to use the flying machine sold to the Government, but does not contemplate the exclusive purchase of patent rights for duplicating the flying machine.

12 Bidders will be required to furnish with their proposal a certified check amounting to ten per cent of the price stated for the 40-mile speed. Upon making the award for this flying machine these certified checks will be returned to the bidders, and the successful bidder will be required to furnish a bond, according to Army Regulations, of the amount equal to the price stated for the 40-mile speed.

13 The price quoted in proposals must be understood to include the instruction of two men in the handling and operation of this flying machine. No extra charge for this service will be allowed.

14 Bidders must state the time which will be required for delivery after receipt of order.

JAMES ALLEN
Brigadier General, Chief Signal Officer of the Army

SIGNAL OFFICE,
Washington, D. C., December 23, 1907

APPENDIX NO. 2

SIGNAL CORPS SPECIFICATION, NO. 483.

Advertisement and Specification for a Dirigible Balloon.

Bidders are requested to read carefully every paragraph of this specification and include in their proposal every detail called for.

To the public.—Sealed proposals, in duplicate, will be received at this office until 12 o'clock noon on February 15, 1908, and no proposals will be considered which are received after that hour.

Persons wishing to submit proposals under this specification can obtain the necessary forms and envelopes by application to the Chief Signal Officer, United States Army, War Department, Washington, D.C. The United States reserves the right to reject any and all proposals.

Unless the bidders are also the manufacturers of the dirigible balloon they must state the name and place of the maker.

Preliminary.—This specification covers the construction of a dirigible balloon, to consist of a gas bag supporting a suitable framework on which will be mounted the necessary propelling machinery.

Inspection.—The Chief Signal Officer of the Army will reserve the right to inspect any and all processes of manufacture, and unsatisfactory material will be marked for rejection by the inspectors before assembling.

Acceptance.—The dirigible balloon will be accepted only after a trial flight, during which it will comply with all requirements of this specification.

GENERAL REQUIREMENTS.

The general dimensions of the dirigible balloon will be determined by the manufacturer, subject to the following conditions:

1 The gas bag shall be designed for inflation with hydrogen. The material for the gas bag shall be furnished by the bidder, and shall be subject to approval by the Chief Signal Officer of the Army, and must have a minimum breaking strength of not less than 62½ pounds per inch width and must require no varnish. The dimensions and shape of the gas bag will be as desired by the manufacturer, except that the length must not exceed one hundred and twenty (120) feet.

2 Inside the gas bag there will be either one or two ballonets having a total capacity of at least one-sixth the total volume of the gas bag. Leading to the ballonets there will be tubes of proper size connected to a suitable centrifugal blower for maintaining a constant air pressure in the ballonets. The approved fabric for the ballonets must have a minimum tensile strength of not less than 48½ pounds per inch width.

3 *Valves.*—In the lower part of the ballonet and gas bag, or on the ballonet air tubes near the gas bag, there will be an adjustable automatic valve designed

to release air from the ballonet to the outside atmosphere. On the under side of the gas bag there will be a second adjustable automatic valve of suitable size, so designed as to release hydrogen from the interior of the gas bag to the outside atmosphere. This valve will also be arranged so that it may be opened at will by the pilot.

4 In the upper portion of the gas bag there will be provided a ripping strip covering an opening five (5) inches wide by six (6) feet long, with a red rip cord attached in the usual manner and brought down within reach of the pilot through a suitable gas-tight rubber plug inserted in a wooden ring socket.

5 The suspension system and frame must be designed to have a factor of safety of at least three, taking into account wind strains as well as the weight suspended.

6 A type of frame which can be quickly and easily assembled and taken apart will be considered an advantage.

7 The balloon must be designed to carry two persons having a combined weight of 350 pounds; also at least 100 pounds of ballast, which may be used to compensate for increased weight of balloon when operating in rain.

8 The dirigible balloon should be designed to have a speed of twenty miles per hour in still air, but bidders must submit quotations in their proposals for cost depending upon the speed attained during the trial flight acording to the following schedule:

20 miles per hour, 100 per cent.
19 miles per hour, 85 per cent.
18 miles per hour, 70 per cent.
17 miles per hour, 55 per cent.
16 miles per hour, 40 per cent.
Less than 16 miles per hour rejected.
21 miles per hour, 115 per cent.
22 miles per hour, 130 per cent.
23 miles per hour, 145 per cent.
24 miles per hour, 160 per cent.

9 The speed accomplished during the trial flight will be determined by taking an average of the time over a measured course of between two and five miles against and with the wind. The time will be taken by a flying start, passing the starting point at full speed at both ends of the course. This test subject to such additional details as the Chief Signal Officer of the Army may prescribe at the time.

10 Provision must be made to carry sufficient fuel for continuous operation of the engine for at least two hours. This will be determined by a trial endurance flight of two hours, during which time the airship will travel continuously at an average speed of at least 70 per cent of that which the airship accomplishes during the trial flight for speed stated in paragraph 9 of this specification. The engine must have suitable cooling arrangements, so that excessive heating will not occur.

11 Three trials will be allowed for speed as provided for in paragraph 9, and three trials for endurance as provided for in paragraph 10, and both tests must be completed within a period of thirty days from the date of delivery, the expense of the tests to be borne by the manufacturer. The place of delivery to the Government and trial flights will be at Fort Myer, Virginia.

12 The scheme for ascending and descending and maintaining equilibrium must be regulated by shifting weights, movable planes, using two ballonets or other approved method. Balancing by the aëronaut changing his position will not be accepted.

13 This dirigible balloon will be provided with a rudder of suitable size, a manometer for indicating the pressure within the gas bag, and all other fittings and appurtenances which will be required for successful and continuous flights, according to this specification.

14 Bidders will be required to furnish with their proposal a certified check amounting to fifteen per cent of the price stated for the 20-mile speed. Upon making the award for this airship these certified checks will be returned to bidders, and the successful bidder will be required to furnish a bond, according to Army Regulations, of the amount equal to the price stated for 20-mile speed.

15 Bidders must submit with their proposals drawings to scale showing the general dimensions and shape of the dirigible balloon which they propose to build under this specification; the horsepower and description of the engine which will be used for the motive power; the size, pitch and number of revolutions of the propellers; drawing illustrating the suspension system for attaching frame to gas bag; horse power and description of blower for forcing air into ballonets; volume of gas bag; volume of ballonets; the material of which the frame will be constructed; size of valves, etc. Plans received will not be shown to other bidders.

16 Bidders must furnish evidence that the Government of the United States has the lawful right to use all patented devices or appurtenances which may be part of the dirigible balloon and that the manufacturers of the dirigible balloon are authorized to convey the same to the Government. This refers to the right of the Government to use this dirigible balloon without liability for infringement of other inventors' patents. It does not contemplate the exclusive purchase of patent rights for duplicating the airship.

17 The prices quoted in proposals must be understood to include the instruction of two men in the handling and operation of this airship. No extra charge for this service will be allowed.

James Allen
Brigadier General, Chief Signal Officer of the Army

Signal Office
Washington, D. C., January 21, 1908

Fig. 1 French Dirigible "Patrie"

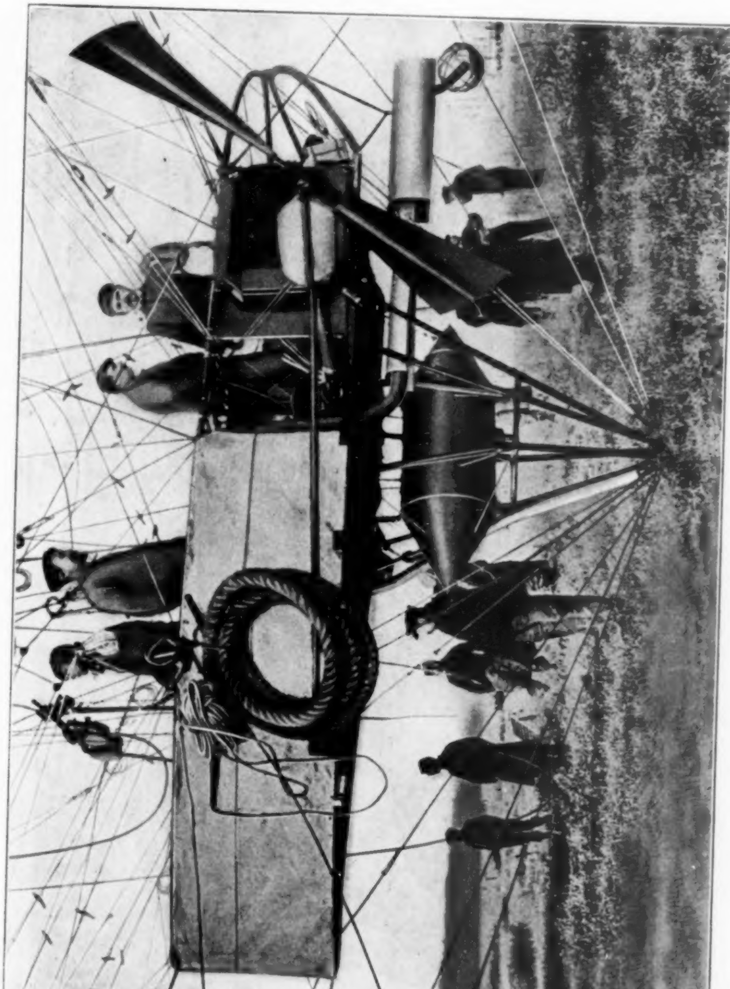

FIG. 2 FRENCH DIRIGIBLE "PATRIE;" DETAILS OF CAR

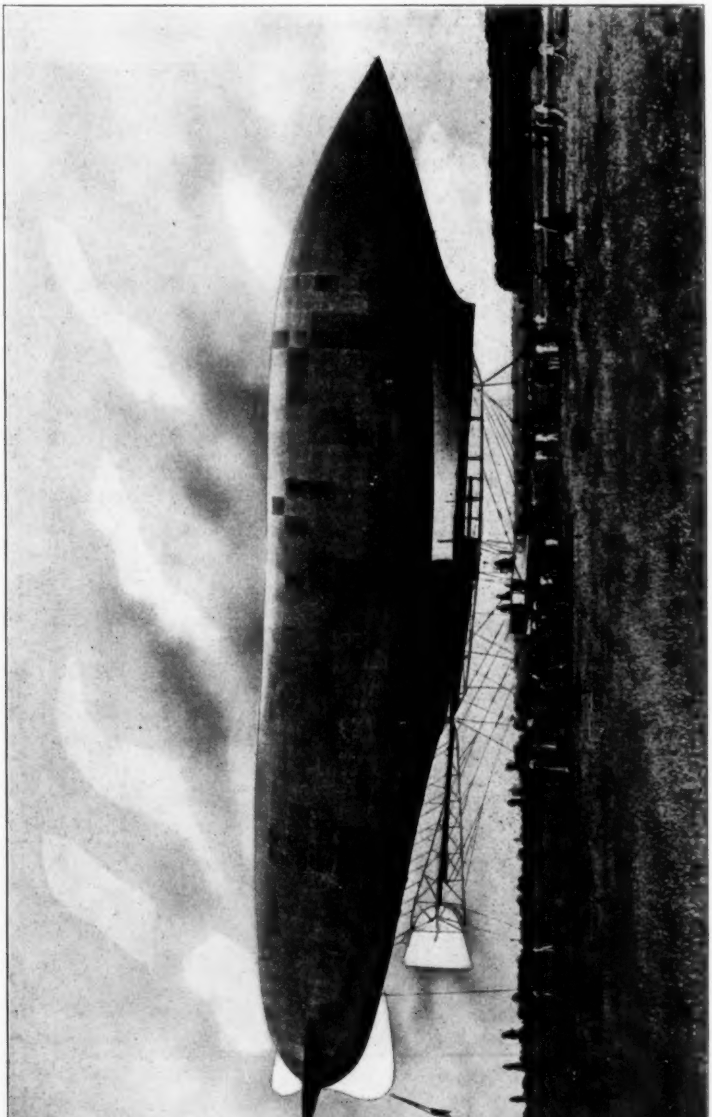

FIG. 3 FRENCH DIRIGIBLE "REPUBLIQUE"

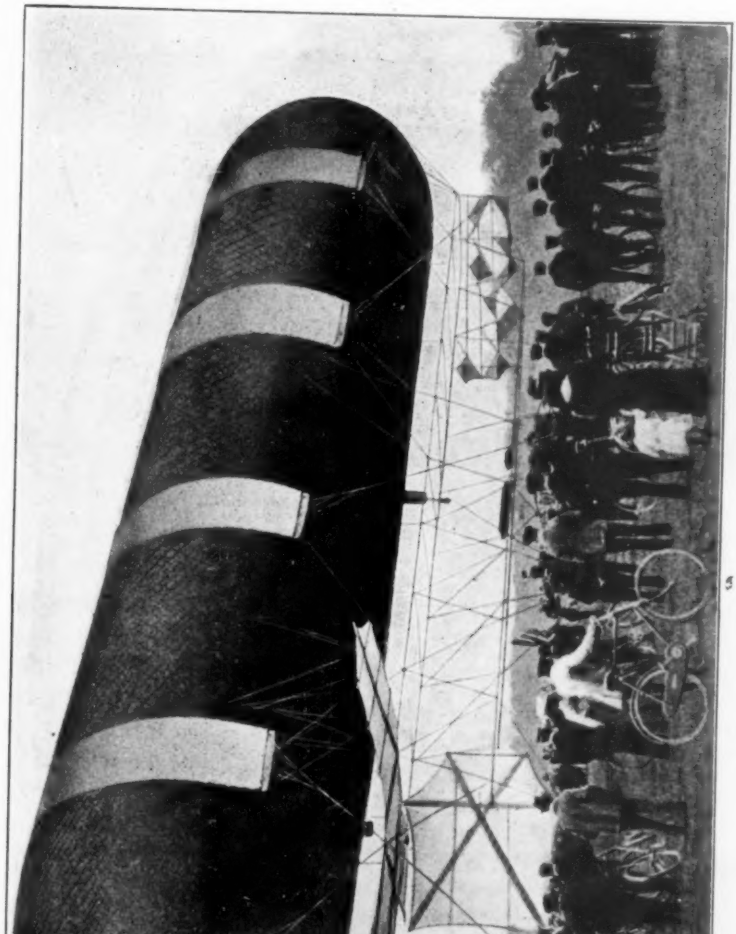

Fig. 5 English Dirigible No. 1

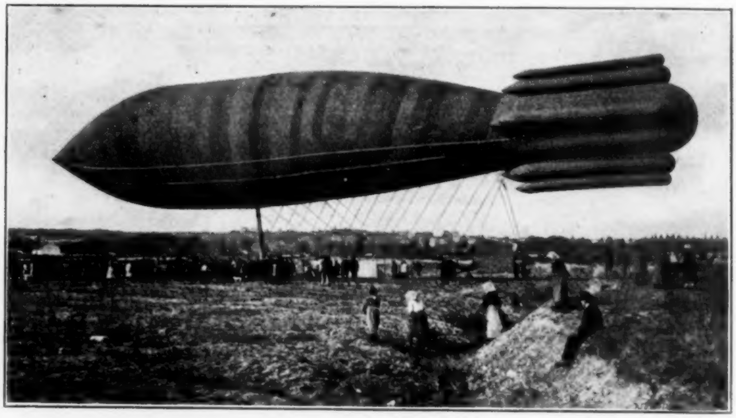

FIG. 4 FRENCH DIRIGIBLE "VILLE DE PARIS"

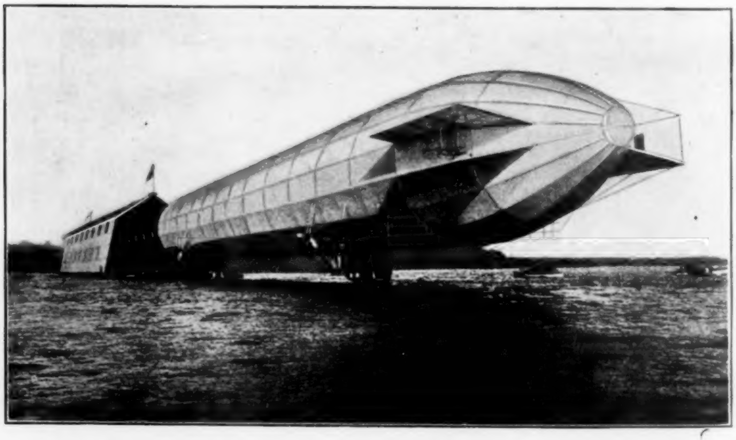

FIG. 6 GERMAN DIRIGIBLE "ZEPPELIN" WITH FLOATING HANGAR

FIG 7 GERMAN DIRIGIBLE "ZEPPELIN;" DETAILS OF CAR

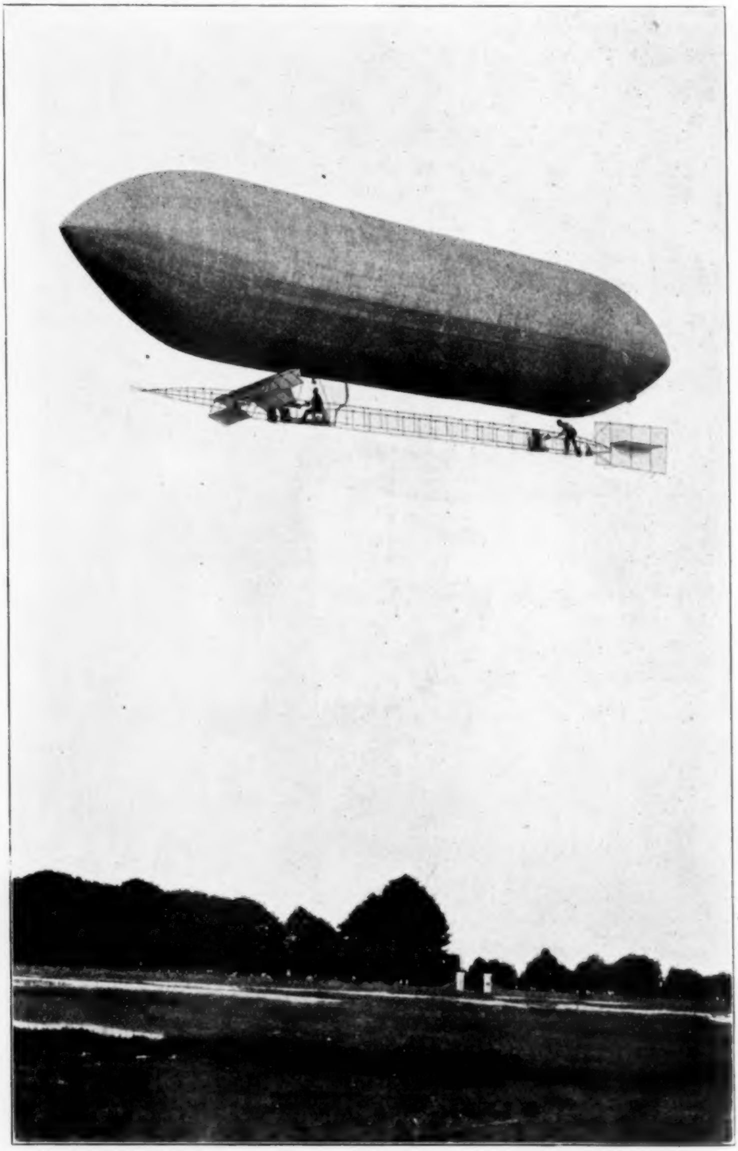

FIG. 8 SIGNAL CORPS DIRIGIBLE NO. 1, IN FLIGHT, FORT MYER, VA., AUGUST 1908

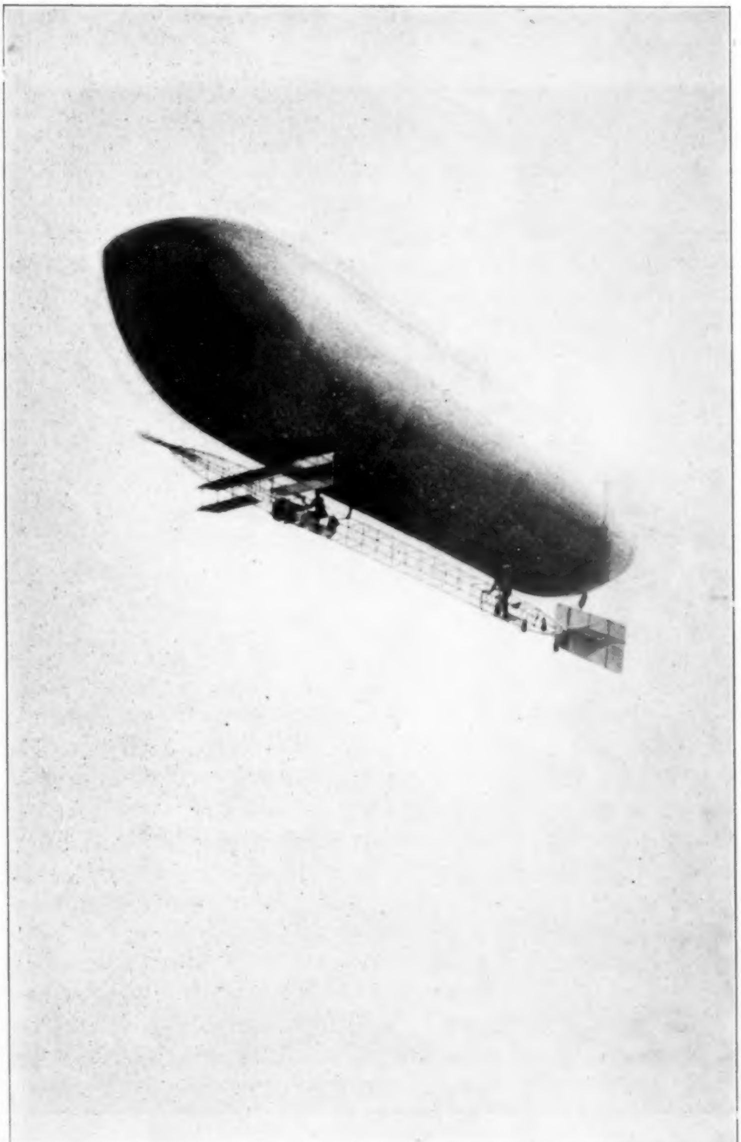

FIG. 9 SIGNAL CORPS DIRIGIBLE NO. 1, IN FLIGHT, FORT MYER VA., AUGUST 1908

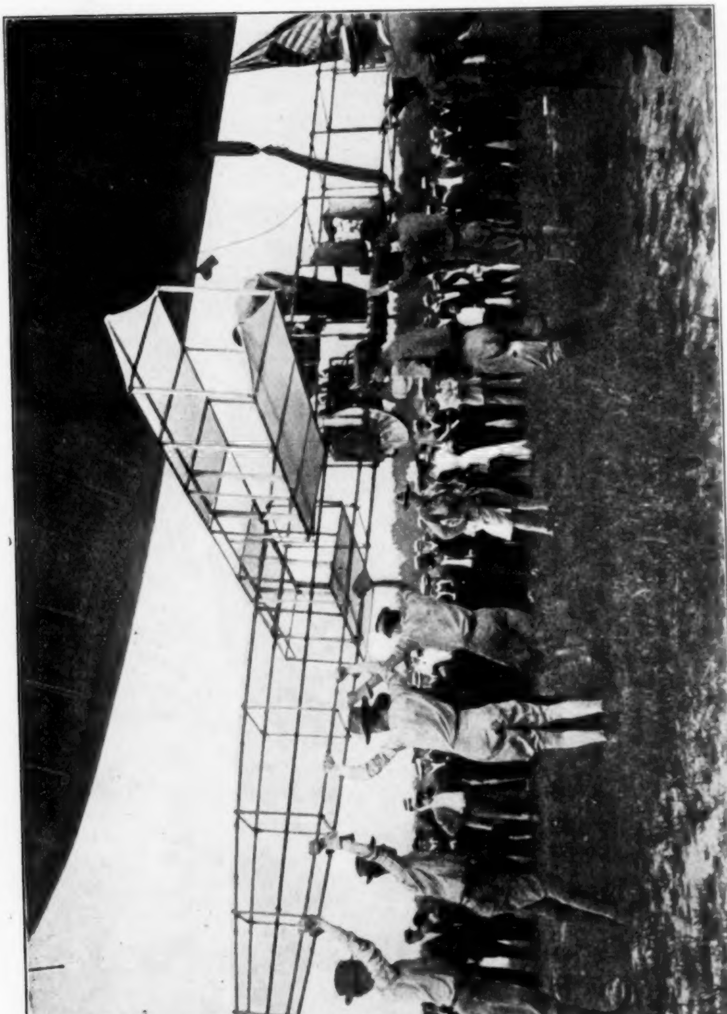

Fig. 10 Signal Corps Dirigible No. 1, Showing Details of Front Maneuvering Planes

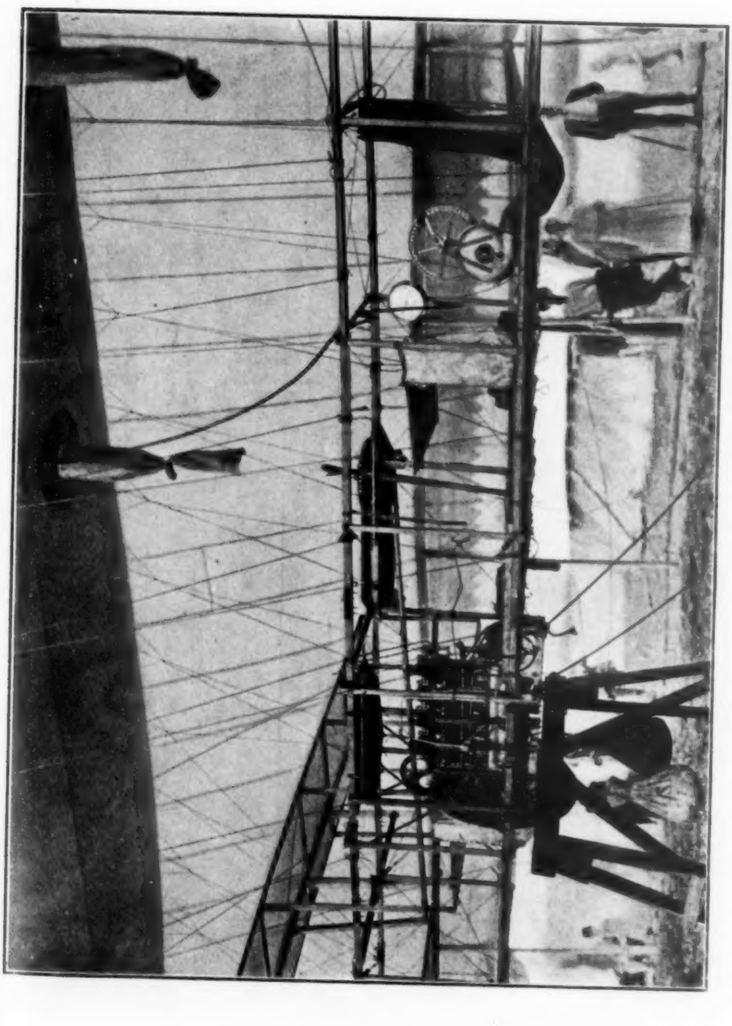

Fig. 11 Signal Corps Dirigible No. 1, Showing Details of Car

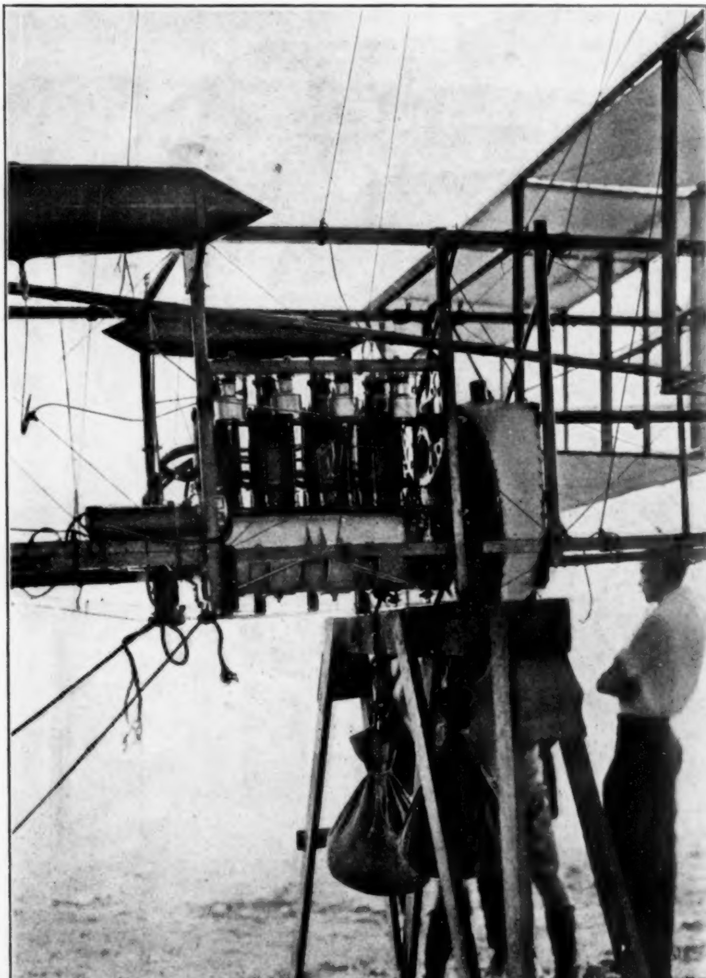

Fig. 12 Signal Corps Dirigible No. 1, Showing Details of Engine

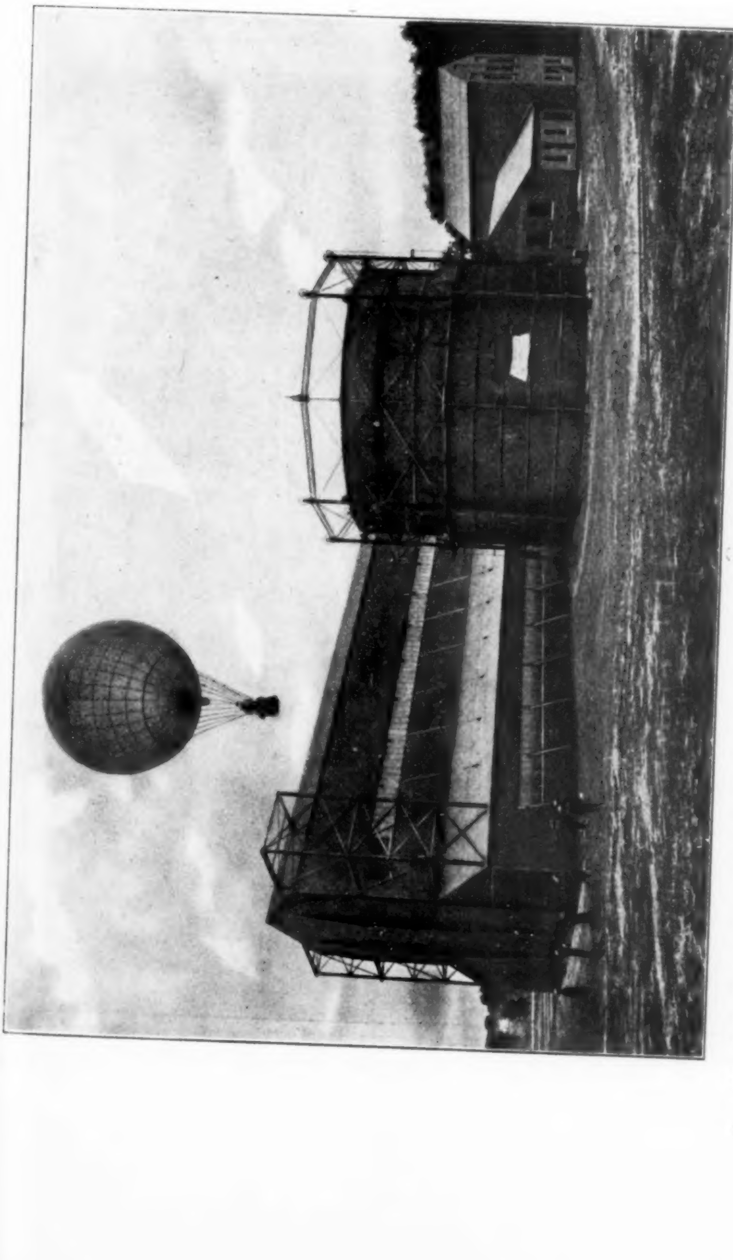

FIG. 13 STEEL BALLOON HOUSE, GASOMETER AND HYDROGEN-GENERATING PLANT, SIGNAL CORPS POST, FT. OMAHA, NEB

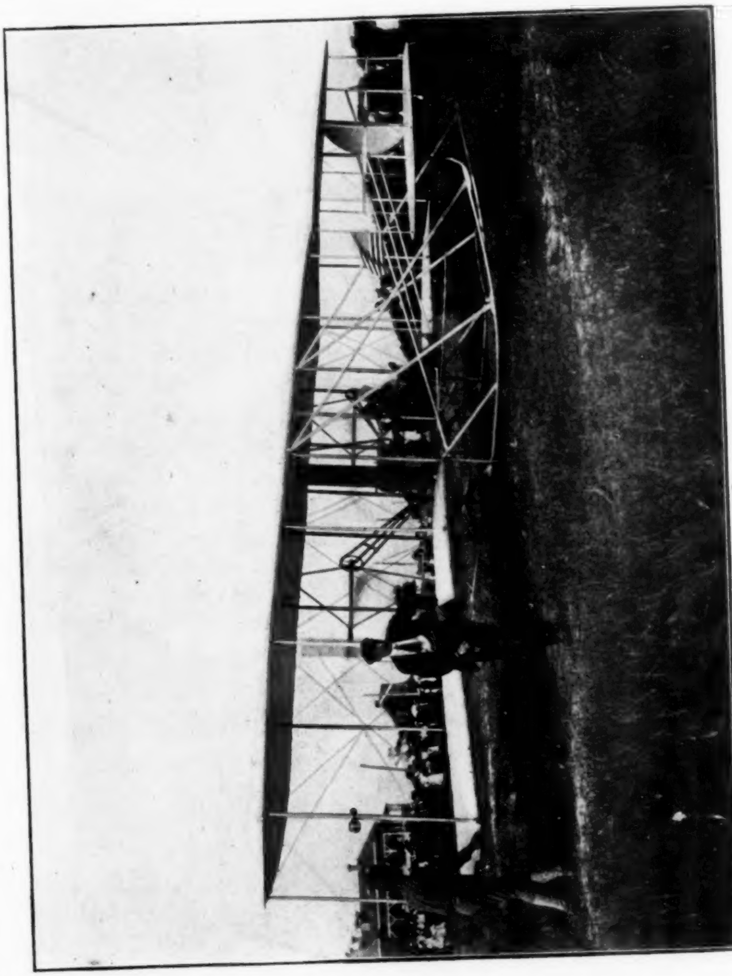

Fig. 14 Wright Brothers' Aëroplane, Fort Myer, Va., September 12, 1908; Ready for the Start; Orville Wright and Passenger

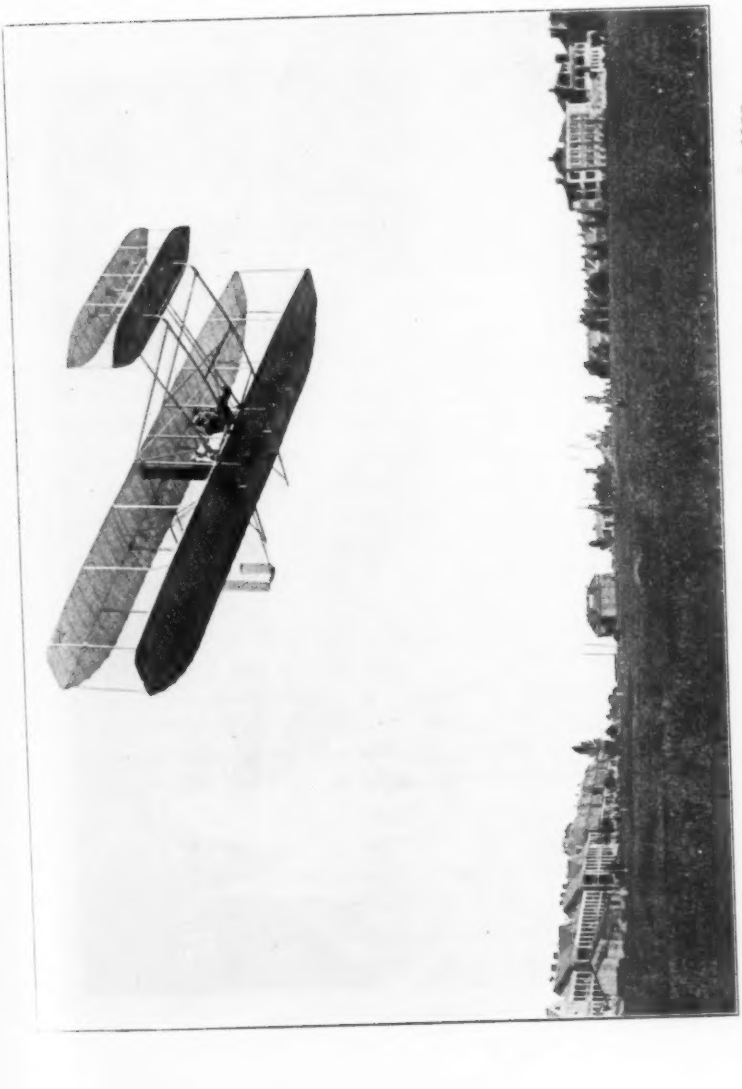

FIG. 15 WRIGHT BROTHERS' AËROPLANE, FORT MYER, VA., SEPTEMBER 9, 1908

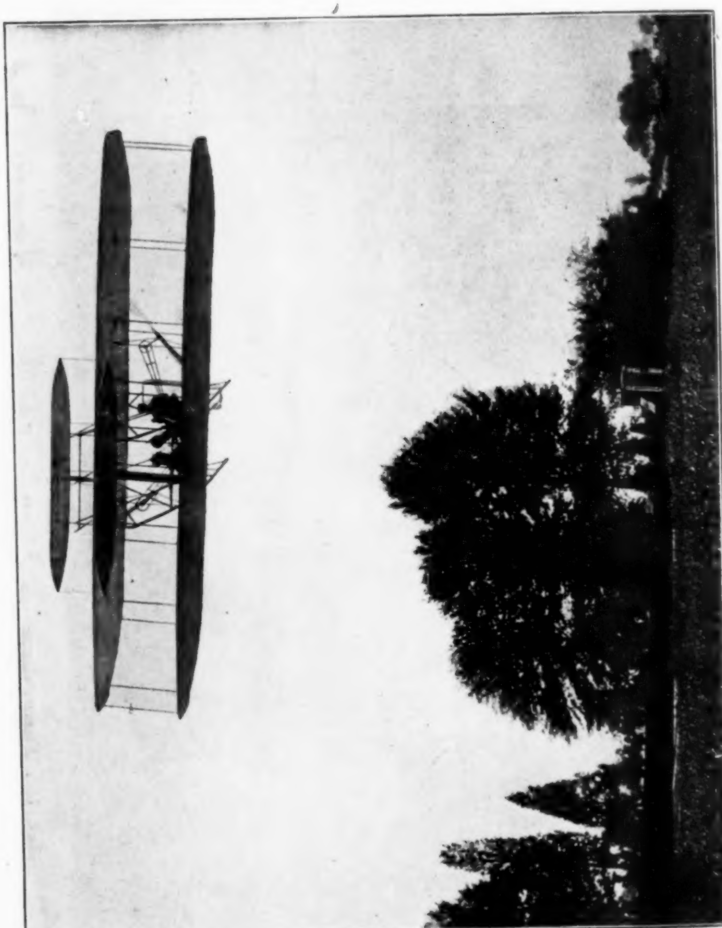

FIG. 16 WRIGHT BROTHERS' AËROPLANE, FORT MYER, VA., SEPTEMBER 9, 1908

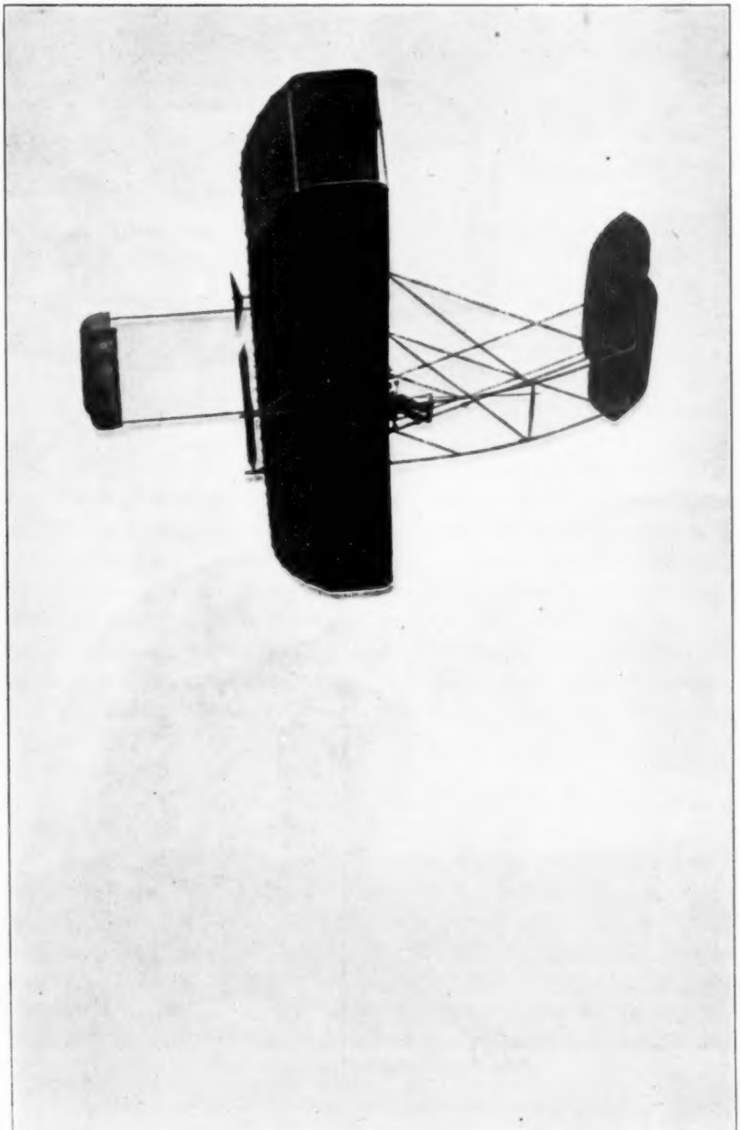

FIG. 17 WRIGHT BROTHERS' AËROPLANE, FORT MYER, VA., SEPTEMBER 12 1908

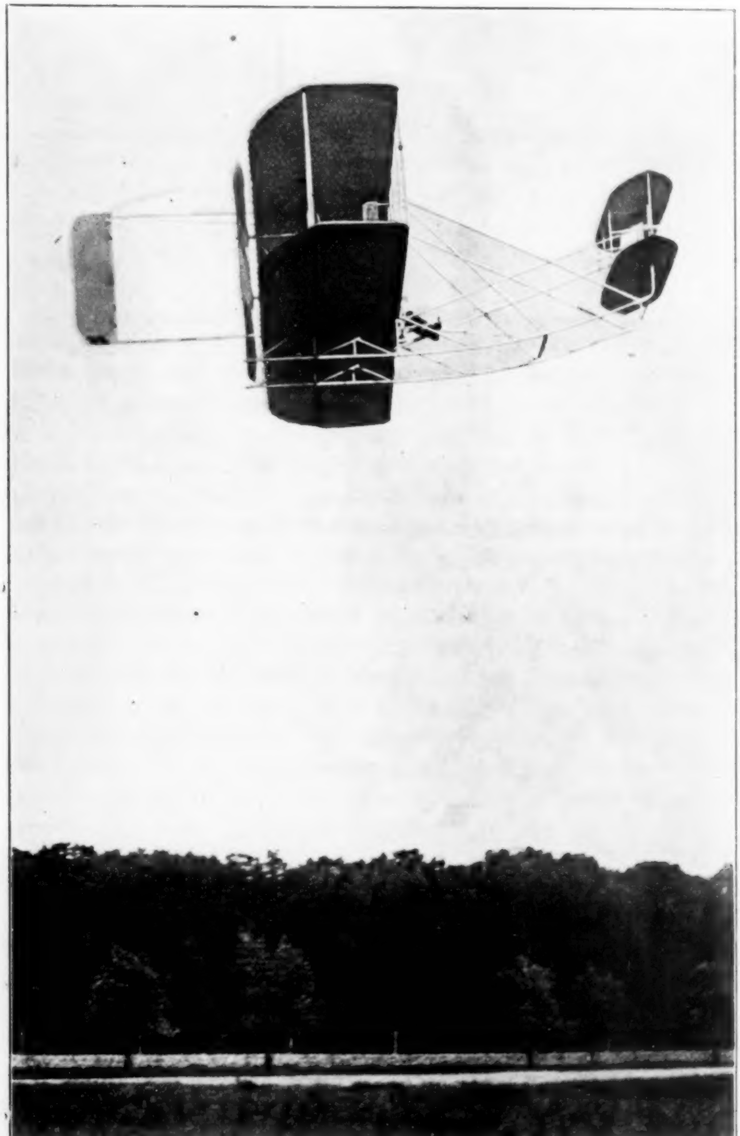

FIG. 18 WRIGHT BROTHERS' AËROPLANE, FORT MYER, VA., SEPTEMBER 12, 1908. TIME OF FLIGHT, 1 HR., 14 MIN., 20 SEC.

FIG. 19 WRIGHT BROTHERS' AËROPLANE, FORT MYER, VA., SEPTEMBER 12, 1908. TIME OF FLIGHT, 1 HR., 14 MIN., 20 SEC.

FIG. 20 WRIGHT BROTHERS' AËROPLANE, FORT MYER, VA., SEPTEMBER 12 1908. ORVILLE WRIGHT AND PASSENGER. TIME, 9 MIN., 6 SEC.

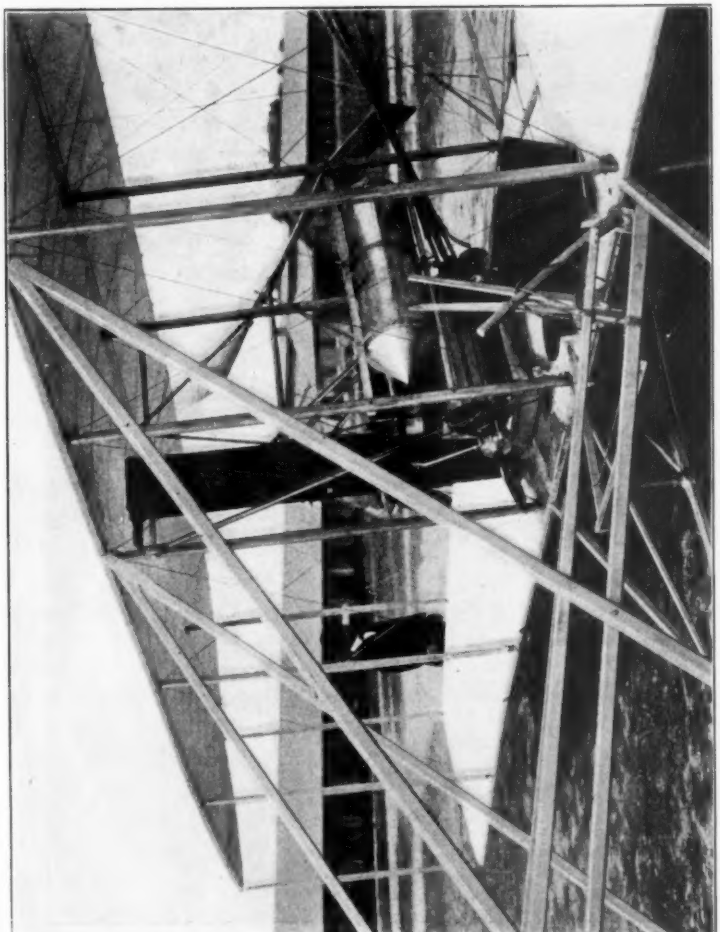

FIG. 21 WRIGHT BROTHERS' AËROPLANE; DETAILS OF CONSTRUCTION

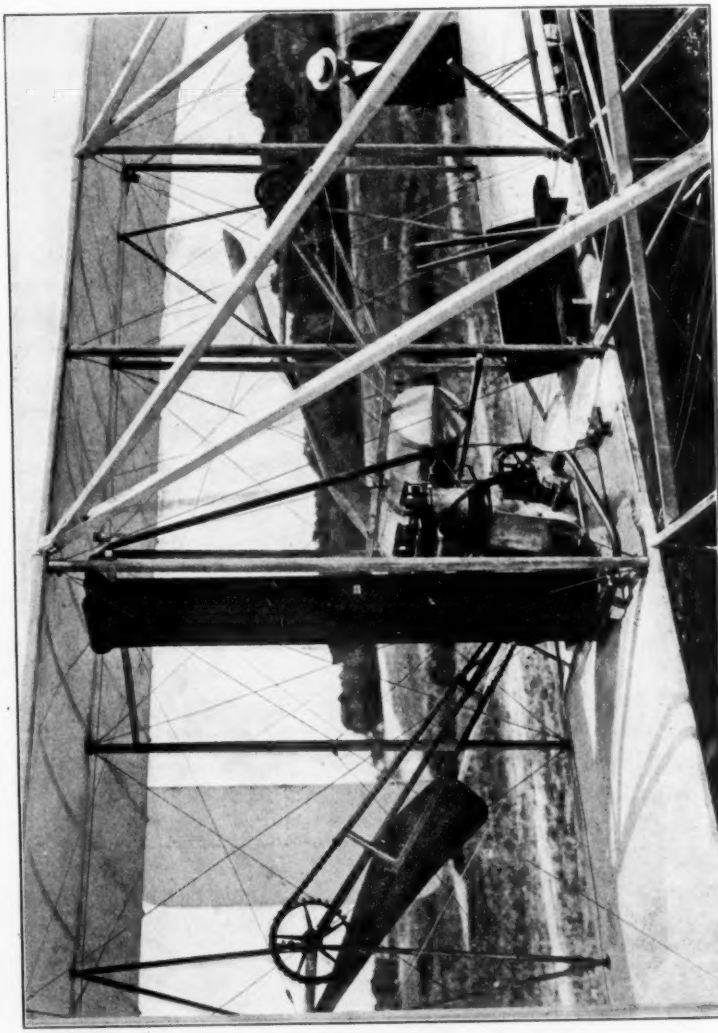

Fig. 22 Wright Brothers' Aroplane; Details of Construction

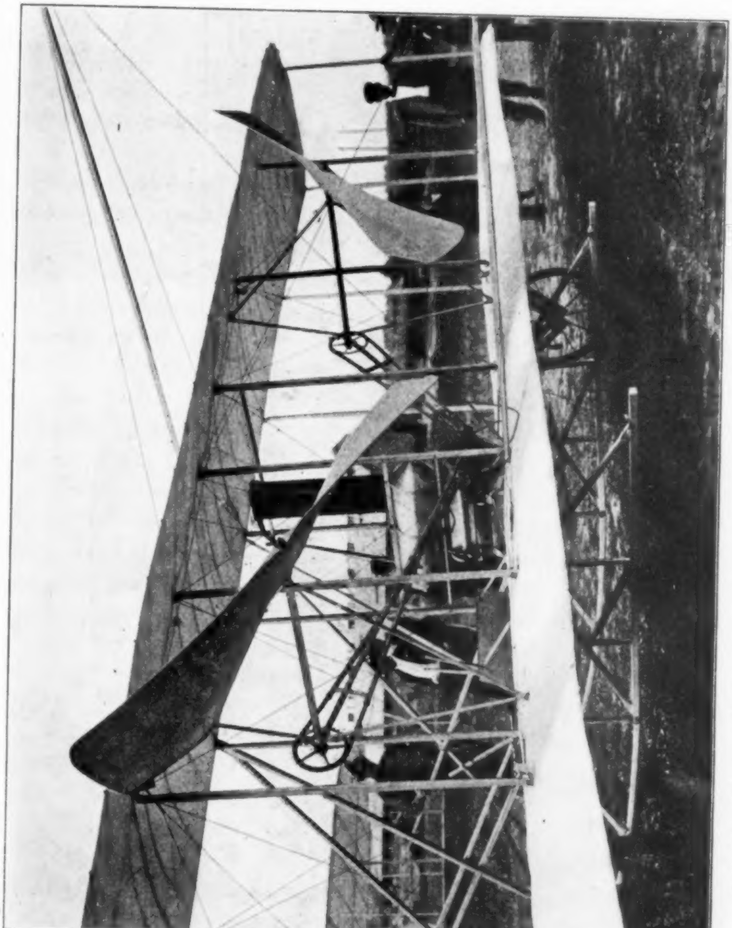

FIG. 23 WRIGHT BROTHERS' AËROPLANE; DETAILS OF CONSTRUCTION, REAR VIEW

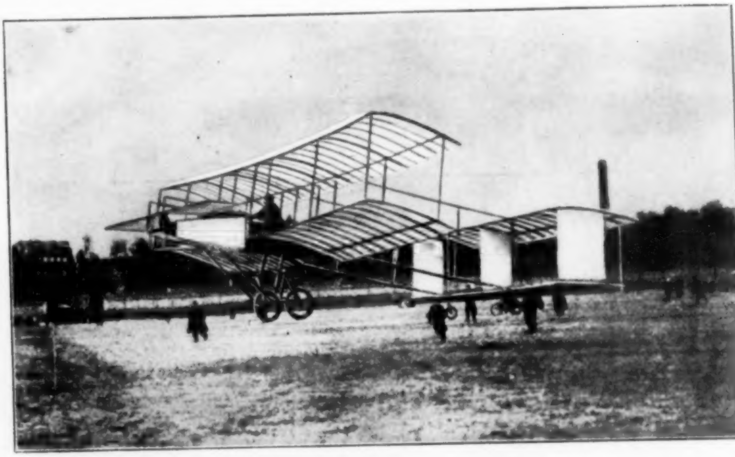

FIG. 24 FARMAN AËROPLANE

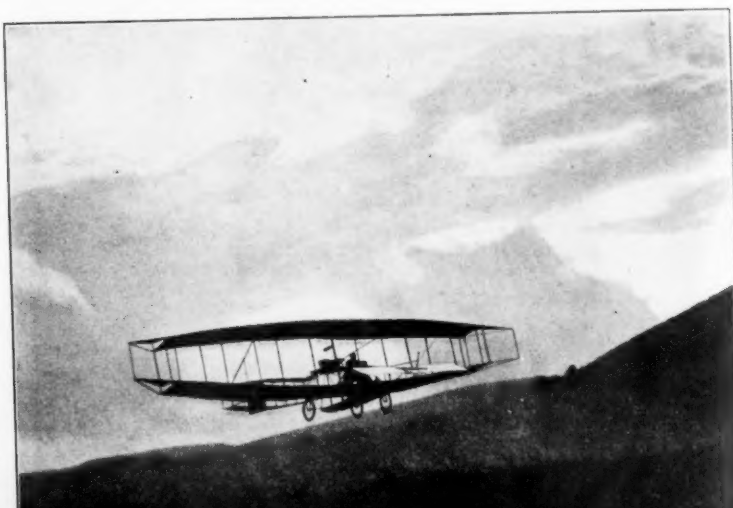

FIG. 25 "JUNE BUG" AËROPLANE, HAMMONDSPORT, N. Y. AËRIAL EXPERIMENT ASSOCIATION

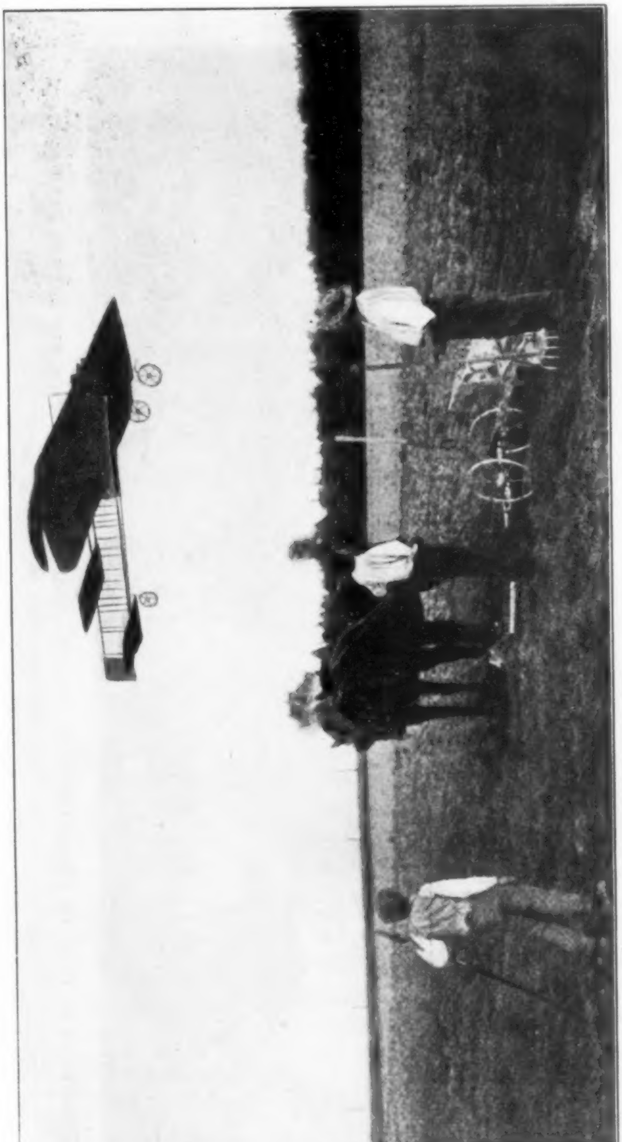

FIG. 26. FROM TOWN TO TOWN BY AËROPLANE. M. BLERIOT PASSING OVER A FARM DURING HIS FLIGHT FROM TOURY TO ARTENAY.

From Town to Town by Aëroplane: Mr. Henry Farman Skirting a Church and Passing Over a Little Village near Rheims

DISCUSSION

Dr. W. J. Humphreys The point I am to discuss primarily is the connection of the winds with flying machines. It will be very important for practical navigators of the air to have maps of the winds. Such maps are issued certainly once a day, and may be had twice a day, or even more frequently, if there be a call for them, as I fancy there will soon be on account of the development of the flying machine. That is evident to every one, but there are one or two points with which perhaps you are not quite so familiar.

2 We find that there is often a great difference in the direction of the winds as we ascend. I have seen the wind change in direction by as much as 30, 45 or even 90 deg., at an elevation of not more than 1000 ft., and in one or two instances I have seen winds differ in direction by as much as 180 deg., at a distance of 1000 ft. above the surface of the earth. It will be necessary to take account of these facts in the navigation of the atmosphere.

3 The winds change greatly in velocity also, at short distances above the surface of the earth. We find by experiment with kites and balloons that near the surface of the earth the air is exceedingly turbulent, after the order of a choppy sea, but at distances of 500 ft. or over, it is more like regular billows, while at an elevation of 1000 ft. the wind is practically steady. Here the aëronaut would neither be in reach of the billows nor run into the ruts and hills and bumps of the invisible air.

4 The direction of the wind can frequently be determined by simply observing the clouds. That will be an aid for the practical navigator, but these changes are different at one place from what they are at another place, and that the practical navigator himself will have to watch. Maps cannot be made for this particular purpose, but what we can get from the map primarily is where the storms are and from what direction the winds will be likely to come for ten or fifteen hours. We have found that around the center of the cyclonic storm, winds will blow in the northern hemisphere counter-clockwise, so knowing the position he is in with reference to the center of the storm, the observer can determine the direction in which the winds will blow. In this section of the country the winds which run at all high in summer always blow practically parallel to the coast line from New York to Boston, while in winter the upper winds are more nearly directed towards the ocean. That may be of importance for the practical navigator to take into consideration.

5 There are many more points of this type that I might bring out, but I believe these are really the salient ones.

PROF. JOHN A. BRASHEAR, who was intimately associated with Professor Langley for three years, in his scientific researches, wished to emphasize the recognition given his work by Major Squier, and thus to pay a debt of gratitude to a man whom he considered one of the greatest physicists, as well as one of the most generous men of this country.

2 Par. 159, on the most advantageous speed and angle of flight, had recalled to him the very large number of experiments made by Professor Langley to determine these two important points. Professor Langley invented a machine, which he called a dynanometer-chronograph, to record all the experiments which he made, and to his honor be it said that he started into the work not for the purpose of building a flying machine, but of determining the law of flight, and he repeated his experiments time and time again and sent nothing into the world of which he was not practically certain; just as in sending out his experiments on the absorption of the atmosphere as a factor in the possibility of life on the earth, he was as particular with the data in regard to the former as with reference to the latter.

3 Professor Brashear referred to Professor Langley's failure in attempting the last flight with his aëroplane, and to the unfortunate publication of unjust reports which did so much to check his career and wreck his life. He said: "I want to add one word more, and that is the sad side of it. We have heard of Fulton's being stoned when he started his steamboat and of the anathemas hurled at the men who have been pioneers in the advancement of the world's knowledge. I went into the Smithsonian Institution just after Mr. Langley's unsuccessful flight on the Potomac. Mr. Langley heard my voice in the outer office and came in to me, taking my hand, with bowed head. 'Mr. Brazier,' as he always called me, 'Come in, I want to talk to you.' Picking up two little pieces of steel broken from an original piece and handing it to me he said, 'Here is what has wrecked my life. My life-work is a failure, this broke, and turned my ship into the Potomac instead of up into the air.' And that man could not be aroused from his lethargy and depression. I said to him, 'Professor Langley, your work in the study of the earth's atmosphere and the possibility of life is enough for one man to do;' but it was no consolation to him.

4 "Let us not forget to say a few kindly words, even in the failure of men, and to remember that a great part of this life and of our success in it is made up of our seeming failures."

Mr. Geo. L. Fowler, who was also associated with Professor Langley, gave a few reminiscences of his work, outlining it as follows:

2 Professor Langley started with the idea of determining the resistance and lifting-power of various types of aëroplanes. He started with the simplest of little instruments, which he projected from the balcony of the Smithsonian Institution down into the larger room beneath. These experiments developed first a motor-driven machine, for which he used the contraction of a rubber band, and then supplemented the work which I believe he had already begun at the Allegheny Observatory, by establishing a whirling table in the lower part of the Smithsonian Institution by which he measured the resistance and lifting-power of the wings of various types of birds and almost every conceivable shape and form of aëroplane, and also obtained the propelling-power of various types of propellers fit for use in the air. On one occasion he built a large propeller mounted on a hand-car which he used on the Pennsylvania Railroad, and obtained possession of a small track on which he drove that car for several days in order to obtain the resistance of the car and also the propelling-power of the machine.

3 I think it was only after the first successful flight of his motor-driven aëroplane that Professor Langley realized the importance of balance. Any one who has seen Orville Wright drive his machine at Fort Myer must recognize after witnessing it for even a few minutes the absolute control which he seems to possess over the machine, and if any here had been present at one of the flights, and had been invited to take a ride with Mr. Wright, they would have gone with perfect confidence. The machine started down a slight incline or monorail, rose, and apparently gave a slight dip—it looked to me as though he had his upper wings elevated to give him the rising power, found himself rising too suddenly, turned back and after a slight hesitation flew away. The steering of the machine was under absolutely perfect control. One thing about the Wright machine I think Professor Langley overlooked in his early experiments. In all early work and in the publications of the Smithsonian Institution, the principle of the bird soaring on motionless wings was repeated again and again. Now an eagle soaring high above the earth appears to be on motionless wings; but, if you will watch a buzzard rising over the land, or stand at the stern of an ocean steamer and watch the gulls following it, you will see that they do identically the same thing as a boy walking on a railroad track or a tight rope walker on the rope with a balancing pole—the soaring bird with his wings out is constantly

making sudden, quick motions. That point, which Professor Langley failed to overcome, the Wright Brothers overcame by a slight warp in the upper wing, which has been one of the causes of the success which they have obtained.

4 Professor Langley found on his first experiment that balancing was a great necessity. In his first machine, the weights were carefully calculated in order to determine the exact specific gravity of the whole thing, and then a small hollow tin can four or five inches in diameter and six or seven inches long was placed on the bowsprit to lower the specific gravity to such an extent that the whole machine would float if dropped into the water. When first cast off, the can was pretty well forward, and the machine promptly made a dive head-foremost down into the Potomac. Next the can was moved back four or five inches, throwing the center of gravity back so much further that the machine went up into the air and dropped back tail-foremost into the water. Then the can was again shifted to a point about halfway between the two, and when cast off the third time it swung around in a circle and flew off for a distance of three-quarters of a mile. That was the first successful flight, and the beginning of the application of balance, the self-limiting balance.

5 In his first experiments Professor Langley worked along the lines of least resistance and lowest weight. Everything about the aëroplane was of the lightest possible type; what he wanted was to fly, and the consideration of stress of material would come later; as a matter of fact in calculating his stresses he ran them up almost to the breaking-point of the material, so that there was practically no factor of safety, and perhaps this was one of the causes of his later failure.

6 I remember once a gentleman saying to him, in discussing his boiler, "Well, Professor Langley, that boiler will make steam, but it will be very uneconomical in the use of fuel." "Never mind that, we will burn gold under the boiler, if necessary. What we want is steam," was the reply. The result on the first successful machine was a very great steam producer, but at the same time a very uneconomical boiler in the use of fuel. His engine was of the simplest possible type, a little engine with a cylinder of about 33 mm. diameter, 73 mm. in stroke. Everything about the work was done on the metric system. He succeeded in developing about one horse power. The cylinder of the engine was a small steel tube, about one millimeter thick, and as the steel would not make a good bearing surface for the piston to move backwards and forwards in, he bushed it with a cast iron bushing of the same thickness, which gives some idea of the delicacy of his work.

The pump that he used to force circulation in the boiler was of the same small and delicate type. The only thing about the machine really efficient from the standpoint of the steam engineer was his burner. He realized that he must burn his fuel to get as high a temperature as possible, and spent a great deal of time in developing his burner, which was exceedingly efficient, and with that small machine of about 14 ft. span of wings, he succeeded in making a successful preliminary flight and the machine did repeat this flight on several occasions afterwards.

7 That is a brief outline of what he attempted to do and what he really succeeded in doing, but beyond that, the science of air dynamics has probably been worked in more thoroughly by Professor Langley than by any other man, and it is doubtful if any one will ever devote so much time and energy to the solution of a scientific problem that promises so little in the way of practical results, as did Professor Langley when he undertook to find out the true theory of flight.

No. 1211

A METHOD OF OBTAINING RATIOS OF SPECIFIC HEATS OF VAPORS

By A. R. Dodge, Schenectady, N. Y.

Member of the Society

The general method of using the throttling calorimeter to determine the average value of specific heat is to supply the high pressure side with steam from which practically all moisture has been eliminated. Neglecting radiation:

$$H_1 = H_2 + C_{p_2} (T_2 \text{ sup} - T_2 \text{ sat})$$

where H is the total heat; T the temperature in degrees fahr.

2 Values determined in this way by several authorities show an increase of C_p with increasing superheat, but as has already been pointed out by others[1] the values of total heat given in the steam tables have been found too unreliable to permit of accuracy in such determinations.

3 The following method, using the throttling calorimeter for determining the ratios of the specific heats at various pressures and superheats, has not involved the use of the steam tables, and is therefore not open to errors from that source.

4 Instead of keeping the pressure and the temperature on the high pressure side of the calorimeter constant, both pressures are kept constant and both temperatures are allowed to vary, the steam always remaining superheated, thereby avoiding errors due to moisture when the steam is assumed to be dry and saturated in its initial condition.

[1] Peake: Proceedings of Royal Institute, June 28, 1905, p. 201; Denton: Stevens' Indicator, October 1905, p. 383.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

5 In the experiments of Grindley, Griessmann and others, steam assumed to be dry and saturated at a pressure P denoted by A is more and more throttled, the various low side data being represented by the curve AA' (Fig. 1). A second test gives another curve BB'.

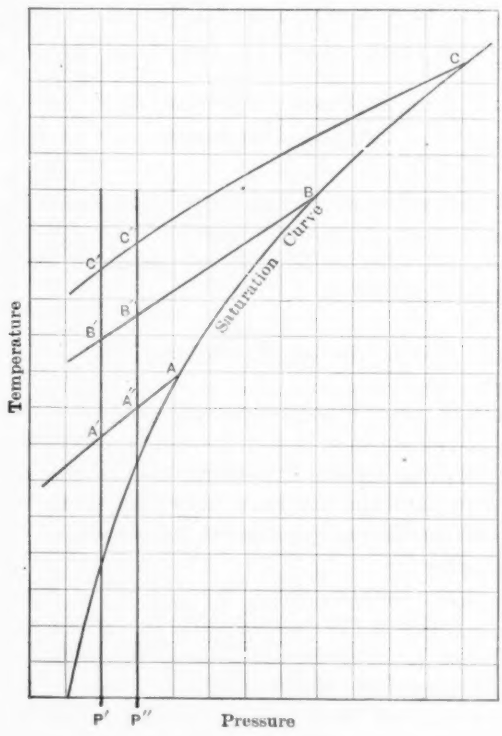

Fig. 1 Old and New Methods of Determination of Variations in Specific Heat

Each of these is a curve of constant total heat, and the value of the total heat of either of them is that of saturated steam at the corresponding point A or B.

6 The difference between the total heats at A' and at B' can then be divided by the temperature difference $A'B'$ to give C_p.

7 The author, on the other hand, finds a number of *pairs* of points, $A'A''$, $B'B''$, $C'C''$, etc., during a run, and uses the points A' and B'

instead of A and B to determine the difference in total heat between A'' and B''. In this way the use of a steam table and the assumption of dry steam are avoided, but the results of the test must be expressed in terms of the values of C_p at the standard pressure P', which was usually 15 lb. per square inch. In other words ratios are obtained, instead of absolute values.

8 The fact that it is very difficult to make sure that the steam is really dry to start with, makes this method much better than the old, even if there were no errors in the total heats given in the steam table.

9 The two pressures being constant through a run, the only variables are the high side temperatures ($A''P''$, $B''P''$, etc.) and the low side temperatures ($A'P'$, $B'P'$, etc.). If the first of these are plotted

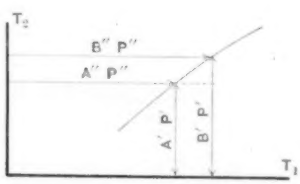

Fig. 2 Graphical Method of Locating Line, the Slope of which Indicates Variations in Specific Heat

as abscissae and the second as ordinates, a smooth curve, Fig. 2, can be drawn which is characteristic of the two pressures. The slope of a line drawn between any two points of this curve is

$$\frac{B' P' - A' P'}{B'' P'' - A'' P''} \qquad [1]$$

Now from Fig. 1, between the temperatures A'' and B'', the average specific heat

$$C_{p_2} = \frac{\text{total heat}}{\text{temp.}} = \frac{H_B - H_A}{B'' P'' - A'' P''}$$

and between the temperatures A' and B' the average specific heat

$$C_{p_1} = \frac{\text{total heat}}{\text{temp.}} = \frac{H_B - H_A}{B' P' - A' P'}$$

As the total heats are equal

$$\frac{C_{p_2}}{C_{p_1}} = \frac{B' P' - A' P'}{B'' P'' - A'' P''} \qquad [2]$$

which is the same value as that found for the slope in equation 1. Therefore variations of curvature shown in Fig. 2, or variations in the tangent of the angle between a line connecting two points of the curve and the horizontal, are a measure of the variations in the specific heats between two pressures at varying superheats.

10 In this way a series of curves for different initial pressures in the calorimeter can be derived, showing the variation of C_p with relation to a standard pressure. The characteristic line Fig. 6, for 15 lb. pressure must pass through the origin at 45 deg., for as there is no pressure change in the calorimeter under this condition,

$$\frac{C_{p_2}}{C_{p_1}} = \text{tangent } \alpha = 1$$

DESCRIPTION OF APPARATUS

11 The general arrangement of boilers, calorimeter, etc., used in these tests is shown in Fig. 3 and 4. This calorimeter was designed after two years of experimental work, during which improvements were made to eliminate radiation, conduction, thermometer errors, temperature lag, and errors in temperature due to velocity of the steam jet. The water was forced by a high pressure pump into two flash boilers connected in series, by means of which the superheat could be raised to any desired amount.

12 The steam was then passed to the separator drum *S* at which the temperature was controlled by injecting water. The pressure and temperature entering the high pressure side *B* of the calorimeter were held constant with practically no variation.

13 The low pressure side of the calorimeter was jacketed, and the steam in the jacket was held at nearly the same temperature as the exhaust from the calorimeter by means of a superheated steam supply at atmospheric pressure. The steam passing through the calorimeter was weighed after being condensed, and Fig. 5 shows the relation between the absolute pressure on the high side and the steam flow.

14 Orifices of two sizes were used, the diameters of which were 0.052 in. and 0.1 in. The object of using the larger orifice was to elimi-

nate the effect of possible radiation, although in addition to jacketing the low pressure chamber, the jacket was thoroughly lagged with magnesia. No different results were obtained, however, when using

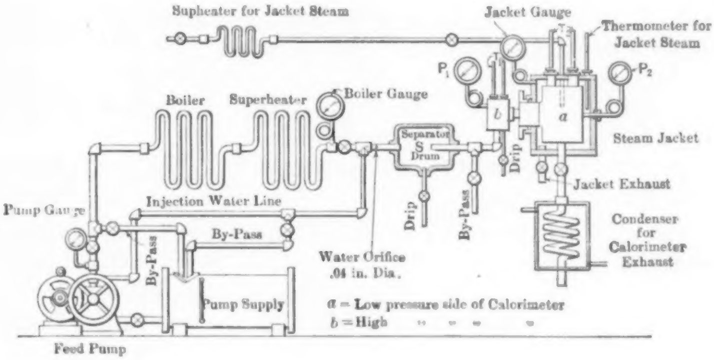

FIG. 3 ARRANGEMENT OF APPARATUS

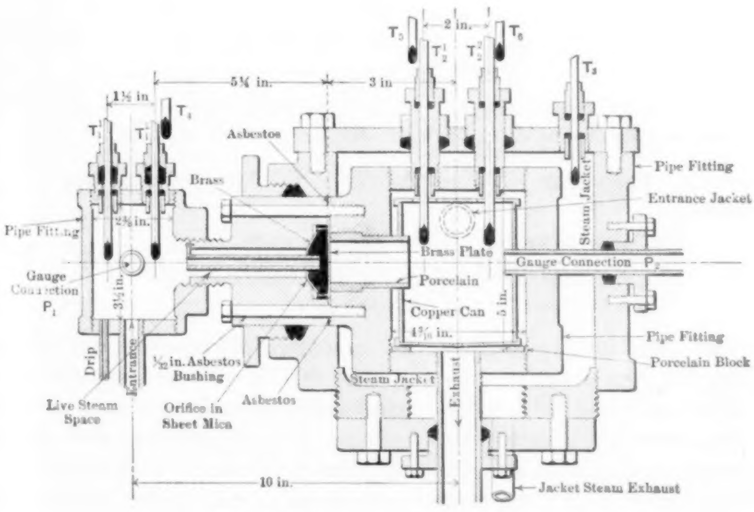

FIG. 4 THROTTLING CALORIMETER

the larger orifice, indicating that radiation was negligible. The question of radiation was further investigated by measuring the drop in superheat in the steam from one chamber of the calorimeter to the other, with the same steam flows as during the regular tests but with

no drop in pressure, as the orifice was entirely removed from the calorimeter. These tests indicate that there is a slight correction in spite of all precautions, but of such small limits that the ratio of the specific heats is not affected.

15 Each of the varying pressure tests, of which there are 85, was run from six to eight hours, each initial temperature point being held constant for about one hour. All pressures on the high pressure side were held constant by means of spring gages calibrated before and after each test, using a reliable dead weight tester.

16 The pressure on the exhaust side of the calorimeter was held at 15 lb. abs. by means of a U-tube gage filled with mercury. The piping connecting this U-tube to the low pressure chamber was closed at the end and holes were drilled through the pipe to avoid a possible error from the velocity head of the steam jet through the orifice. Tests were also made with 165, 8.12 and 2.97 lb. abs. on the low pressure side.

17 The temperature was held constant by means of a calibrated thermometer on the high pressure side. The stem of this thermometer was packed where it passed through the calorimeter wall, and the bulb placed in direct contact with the steam. Another calibrated thermometer was also used in the high pressure chamber as a check. Stem corrections were applied in accordance with the following formula, which has been fully verified for such conditions:

$$\text{Correction} = (T_1 - T_2)\ (T_1 - T_3)\ 0.000087 \text{ deg. fahr.}$$

where

T_1 = observed reading of the chamber.
T_2 = reading just visible above chamber casing.
T_3 = temperature of exposed stem.

18 The thermometers were frequently calibrated in wet steam supplied to a drum at pressures up to 600 lb. These pressures were determined and held at the correct value by a dead weight testing apparatus. The correct temperature at these pressures was determined by thermometers calibrated by the Bureau of Standards at Washington. The thermometers were also corrected for pressure on the bulb by the subtraction of 1 deg. fahr. for each 100 lb. gage pressure, a correction determined by separate experiments in connection with these tests.

19 The thermometers on the low pressure side were also placed directly in the steam and properly packed to prevent leakage around the stem.

METHOD OF MAKING TESTS

20 Readings were not recorded until the apparatus was thoroughly heated and conditions constant.

21 In order to prevent temperature lag, the initial conditions were held constant for an hour if necessary and frequent readings taken on the lower temperature T_2. The initial temperature could be controlled within one-half a degree of the desired value. There was no variation in the initial and final pressures which were both controlled by throttling. After all conditions had remained constant over a satisfactory period the superheat on T_1 was changed and the operation was repeated, the pressure conditions remaining unchanged.

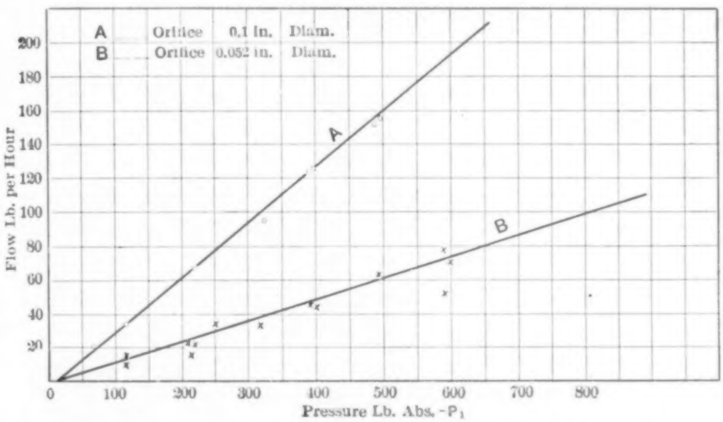

Fig. 5 Steam Flow through Calorimeter at Different Pressures under Average Superheated Conditions

22 The maximum velocity of the steam in the separator drum after the water was injected was never greater than $\frac{25}{100}$ ft. per second; therefore, approximately four seconds elapsed while the steam was passing through the drum, giving the highly superheated mixture sufficient time to become of uniform temperature. The small quantity of water injected, together with the fact that there were three drip pipes between the water orifice and the thermometer on the high pressure side, as well as the location of the thermometer bulbs opposite the jacketed orifice, all indicate that the temperature of the sample admitted to the orifice was correctly observed.

23 In order to investigate the effect of the kinetic energy of the steam jet in the low pressure side, four screens of fine copper gauze

were placed in series in the path of the jet between the orifice and the low pressure thermometers, but the same results were obtained without the screens by raising the thermometers out of the path of the jet.

24 The results of 35 runs made with the apparatus in its final form are shown in the accompanying table. A few characteristic

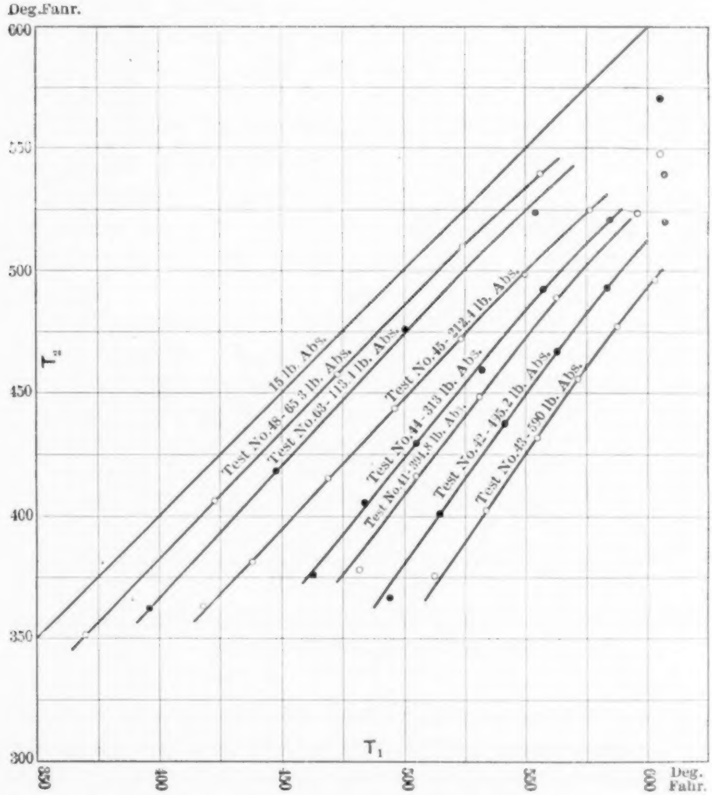

FIG. 6 DIAGRAM SHOWING RELATION BETWEEN TEMPERATURES IN HIGH AND LOW PRESSURE CHAMBERS. THE TANGENTS OF THESE LINES INDICATE THE LAW OF VARIATION OF SPECIFIC HEAT WITH PRESSURE AND TEMPERATURE.

tests have been plotted in curve form, in Fig. 6. The deviation from a straight line is slight, but even this amount would account for a considerable variation in C_p due to the temperature change.

25 The curvature of these lines thus shown graphically is brought out arithmetically in the table, column 11. To avoid magnified

TABLE 1 RESULTS OF 35 RUNS

Test No.	P_1	P_2	T_1sat	T_1	T_2	Jacket T_3	Steam Flow Per Hour	Dia. Orifice In.	Points	Slope $\frac{Cp_1}{Cp_2}$
35	592.9	165.4	485.5	598.1	514.5	505.	82.0	.052		
36	604.8	164.8	487.9	603.7	519.7	507.	78.0	.052		
				580.8	494.5	487.	84.0		1–3	1.190
				560.5	468.3	455.	88.0		2–4	1.259
				538.9	441.8	445.	92.0		3–5	1.297
				512.8	406.4	415.	96.0			
37	511.0	165.0	469.7	604.3	534.0	520.	56.0	.052		
				578.3	507.0	510.	55.0		1–3	1.098
				551.9	476.5	480.	52.0		2–4	1.196
				519.8	437.0	435.	56.0		3–5	1.213
				498.3	411.5	415.	63.0			
38	316.9	164.4	423.5	608.8	555.0	545.	27.0	.052		
				591.0	542.3	530.	27.5		1–3	0.921
				538.0	489.8	490.	29.0		2–4	1.017
				505.8	455.7	460.	29.7		3–5	1.027
				473.0	423.1	430.	30.5			
39	396.0	164.5	444.8	607.2	548.0	535.	38.0	.052		
				569.8	514.5	515.	37.2		1–3	1.050
				505.8	441.4	445.	38.4		2–4	1.150
				473.8	404.2	405.	41.0			
40	398.2	15.0	444.7	605.8	524.0	515.	43.0	.052		
				558.1	477.0	510.	43.5		1–2	0.985
				483.7	387.5	382.	46.0		2–3	1.202
41	394.8	15.0	443.2	596.2	524.2	510.	44.5	.052		
				563.0	488.0	480.	45.0		1–3	1.167
				531.2	448.3	445.	46.0		2–4	1.249
				505.2	415.8	415.	48.5		3–5	1.400
				481.3	378.5	380.	49.5			
42	495.2	15.0	466.2	606.6	519.5	515.	56.0	.052		
				584.1	492.8	490.	57.0		1–3	1.225
				562.5	465.5	450.	57.5		2–4	1.301
				541.3	437.1	430.	57.7		3–5	1.373
				515.2	400.5	400.	62.0		4–6	1.500
				494.2	366.5	370.	64.0			
43	590.0	15.0	485.8	603.3	495.5	485.	75.0	.052		
				587.8	476.5	473.	77.5		1–3	1.312
				572.4	454.9	445.	79.0		2–4	1.382
				555.5	431.8	430.	80.0		3–5	1.376
				533.9	401.9	415.	83.0		4–6	1.331
				512.8	374.9	385.	85.0			
44	313.0	15.0	422.0	606.7	540.0	525.	32.5	.052		
				585.0	520.0	510.	33.0		1–3	1.020
				558.7	491.0	485.	33.5		2–4	1.174
				531.8	457.5	450.	34.0		3–5	1.163
				505.6	429.2	420.	34.5		4–6	1.109
				484.3	404.9	415.	35.5		5–7	0.988
				462.1	376.3	365.	36.0			

TABLE 1 RESULTS OF 35 RUNS—Continued

Test No.	P_1	P_2	T_1sat	T_1	T_2	Jacket T_2	Steam Flow Per Hour	Diameter Orifice	Points	Slope $\frac{Cp_1}{Cp_2}$
45	212.4	15.	388.0	604.5	548.0	525.	19.0	.052		
				576.5	525.0	505.	21.2		1–3	0.927
				549.5	497.1	490.	21.6		2–4	1.006
				522.8	471.0	455.	22.0		3–5	1.020
				496.0	442.5	445.	22.2		4–6	1.028
				469.3	415.0	415.	22.5		5–7	1.063
				438.3	381.1	375.	23.0		6–8	1.022
				418.0	362.6	355.	23.5			
46	112.4	15.	336.0	450.2	409.0	405.	10.0	.052		
				385.9	352.3	345.	10.0			
47	112.2	15.	336.0	601.0	572.3	535.	34.0	.100		
				570.0	549.0	565.	32.0		1–3	0.948
				529.0	504.0	490.	34.0		2–4	1.025
				498.3	475.5	475.	35.0		3–5	1.016
				468.0	442.0	440.	36.5		4–6	1.072
				411.0	381.9	375.	37.5		5–7	1.063
				374.5	342.5	330.	38.5			
48	65.3	15.	298.5	555.5	539.1	535.	20.0	.100		
				524.0	510.1	500.	20.5		1–3	0.990
				473.0	457.4	450.	22.0		2–4	1.019
				421.6	405.8	400.	22.0		3–5	1.033
				370.5	351.4	355.	23.0			
49	216.6	15.		Not worked up				.052		
50	214.7	15.	387.8	602.7	546.7	535.	15.0	.052		
				560.2	510.5	510.	16.5		1–3	0.972
				517.9	464.3	455.	17.0		2–4	1.150
				470.8	407.7	405.	17.5		3–5	1.202
				429.6	358.2	355.	19.0			
51	595.0	15.	486.2	595.0	465.0	465.	51.0	.052		
52	595.3	15.	486.7	599.3	493.3	487.	52.0	.052		
				580.3	468.0	452.	53.0		1–3	1.319
				567.4	451.2	440.	53.0		2–4	1.730
				541.3	400.5	390.	55.0		3–5	1.853
				509.5	343.9	340.	55.0			
53	594.9	15.	487.0	599.0	490.9	490.	45.0	.052	1–2	1.450
				560.8	435.6	425.	47.0		2–3	1.803
				520.1	362.2	360.	50.0		1–3	1.630
54	215.7	15.	388.0	600.0	562.2	550.	65.0	.100		
				501.0	456.0	450.	70.0		1–3	1.110
				459.0	405.5	400.	70.0		2–4	1.208
				417.0	354.6	345.	75.0			
55	215.7	15.	388.5	600.0	562.2	550.	62.0	.100		
				521.0	477.5	470.	63.0		1–2	1.071
				437.5	381.3	375.	67.0		2–3	1.152

TABLE 1 RESULTS OF 35 RUNS—Continued

Test No.	P_1	P_2	T_1sat	T_1	T_2	Jacket T_2	Steam Flow Per Hour	Diam. Orifice in	Points	Slope $\frac{Cp_1}{Cp_2}$
56	495.7	15.5	466.6	555.6	469.0	460.	151.0	.10		
				524.6	425.0	420.	153.5		1–2	1.451
				492.2	373.5	355.	160.0		2–3	1.280
57	215.8	15.0	387.7	605.8	562.7	555.	60.0	.10		
				579.2	536.8	516.	65.0		1–3	0.973
				552.0	510.4	500.	67.5		2–4	1.073
				526.8	480.6	475.	67.5		3–5	1.222
				499.2	445.9	444.	71.0		4–6	1.135
				462.8	408.0	405.	70.0		5–7	1.122
				426.6	364.4	365.	71.0			
58	215.8	15.0	387.1	589.5	546.5	530.	63.0	.10		
				536.0	490.9	483.	65.0		1–3	1.082
				483.7	432.0	425.	68.0		2–4	1.134
				441.3	383.5	380.	69.0			
59	489.8	15.0	465.7	601.6	517.0	500.	150.0	.10		
				575.2	484.6	480.	150.0		1–3	1.284
				538.2	435.5	432.	155.0		2–4	1.388
				518.2	405.5	400.	157.0		3–5	1.521
				501.7	380.0	379.	157.0		4–6	1.592
				480.5	345.5	345.	160.0			
60	392.0	15.0	442.8	540.4	457.5	435.	120.0	.10		
				514.0	425.2	375.	120.0		1–3	1.382
				482.6	377.7	310.	125.0		2–4	1.541
				461.6	344.4	320.	125.0			
61	322.0	15.0	424.4	602.9	545.0	494.	95.0	.10		
				576.8	518.2	485.	95.0		1–3	1.204
				557.0	489.7	475.	96.0		2–4	1.166
				525.5	458.4	430.	97.5		3–5	1.059
				498.9	428.2	420.	98.0		4–6	1.294
				472.7	390.0	365.	98.0		5–7	1.476
				446.6	351.0	349.	5100.0			
62	386.0	15.0	441.8	601.8	533.5	515.	117.0	.10		
				549.6	474.4	455.	135.5		1–2	1.132
				471.9	364.9	360.	140.0		2–3	1.409
63	113.4	15.0	337.4	605.4	569.8	555.	34.0	.10		
				554.0	523.1	514.	35.0		1–3	0.910
				501.1	474.9	460.	35.0		2–4	0.990
				447.8	418.0	390.	35.0		3–5	1.071
				395.5	361.8	355.	35.0			
64	589.7	15.0	488.0	602.9	492.1	481.	52.5	4 openings each .0139		
				581.7	465.0	452.	55.0		1–3	1.372
				555.1	426.4	421.	57.5		2–4	1.400
				533.5	397.4	391.	60.0		3–5	1.168
				502.0	364.4	356.	60.0			

TABLE 1 RESULTS OF 35 RUNS—Continued

Test No.	P_1	P_2	T_1sat	T_1	T_2	Jacket T_3	Steam Flow Per Hour	Diam. Orifice In.	Points	Slope $\frac{Cp_1}{Cp_2}$
								4 openings each		
65b	313.3	2.97	421.3	590.3	518.8	500.0	27.5	.0139		
				533.3	462.3	448.0	27.5		1–3	1.058
				491.5	414.2	407.0	30.0		2–4	1.197
				460.5	375.1	369.0	30.0			
66	315.8	15.00	423.0	612.8	544.5	527.0	20.0	.0139		
				581.6	514.7	501.0	23.0		1–3	1.001
				548.8	480.4	461.0	24.0		2–4	1.096
				516.5	443.3	432.0	25.0		3–5	1.269
				484.6	405.3	398.0	25.0		4–6	1.171
				452.4	368.3	365.0	27.0			
67	315.8	15.00	423.7	621.0	549.4	528.0	22.5	.0139		
				587.3	517.4	493.0	23.0		1–3	1.021
				553.8	480.7	450.0	25.0		2–4	1.029
				521.0	442.5	434.0	25.5		3–5	1.146
				488.5	405.9	393.0	26.0		4–6	1.160
				455.5	366.5	360.0	26.0			
68	316.4	15.00	422.0	618.5	563.7	550.0	97.5	.10		
				583.5	527.0	507.0	99.0		1–3	1.104
				551.0	489.1	461.0	100.5		2–4	1.177
				518.5	450.5	440.0	103.0		3–5	1.219
				486.1	410.0	394.0	102.5		4–6	1.355
				453.3	362.1	352.0	105.0			
69	315.8	8.12	421.5	614.5	558.2	541.0	90.0	.10		
				581.5	523.0	510.0	92.0		1–3	1.120
				549.5	485.3	469.5	95.5		2–4	1.170
				517.0	447.5	441.0	99.0		3–5	1.214
				484.5	406.3	395.0	101.5		4–6	1.372
				452.0	358.2	355.0	105.0			

errors in the derivative, the slope is taken between points not adjacent, the first with the third, the second with the fourth, etc.

26 Each test is considered by itself rather than drawing one curve through points from a series of tests made under the same initial and final pressures.

27 The discrepancy of two or more tests under this latter method is sufficient to obscure the tendency to slight curvature shown by the individual tests.

28 The first column gives the test number, the second and fifth the high side conditions, the third and sixth the low side conditions, the fourth the observed saturated temperature with initially wet

steam, the seventh the jacket temperature, the eighth the steam flow per hour, the ninth the diameter of the steam orifice, and the last column the slope or ratio of C_p under the high side condition to C_p under the low side condition. Values of specific heat cannot be deduced from these tables without definite initial values; with the proper initial values a method is available for obtaining complete curves of specific heat for all conditions.

DISCUSSION

Dr. Harvey N. Davis[1], who had been particularly interested in Mr. Dodge's work because of its possible bearing on his own paper, following it on the program, expressed his gratitude to Mr. Dodge and to the General Electric Company for the opportunity freely afforded him of examining the apparatus used and of studying carefully for some weeks the original records of the observations. He further said:

2 "Mr. Dodge's scheme of keeping the two pressures constant during a run, together with the extremely ingenious graphical method which he has devised for the interpretation of his results, are a notable advance in the technique of throttling calorimetry. Indeed they are among the most original contributions in that field since Professor Peabody's invention of the instrument. I believe that the throttling calorimeter will turn out to be the most sensitive and valuable instrument at our disposal for the investigation of the thermal properties of superheated steam and other vapors. Any future experiments with it ought certainly to be so arranged as to be immediately available for discussion along the lines laid down in this paper. It would, of course, be an advantage, could the arrangement be such as to facilitate also such a discussion as is suggested in the paper on Total Heats.

3 "Anyone using Mr. Dodge's data should, however, be warned of the necessity of making certain corrections in the low-side temperatures. The usual radiation corrections, which can be determined from tests which he made for that purpose, are uniformly small, seldom exceeding 5 deg. fahr. But even with these corrections his observations afford strong internal evidence of some other as yet unexplained error, by reason of which the low-side temperatures still run consistently low, by from 13 to 15 deg. fahr., the amount of the correction being almost independent of the circum-

[1] Professor, department of physics, Harvard University.

stances of the tests. A detailed discussion of these observations will soon appear in another place.

4 "In conclusion, the straight-line law of Mr. Dodge's first paper is obviously only a rough first approximation to the truth, and *has no theoretical significance whatever.* This fact needs emphasis because the temptation to amuse oneself with thermodynamic manipulations based on a simple law of that sort is very strong. It is true that the lines on Mr. Dodge's Fig. 6 are very nearly straight, but it is also true that a very small curvature in them makes a very great difference in any conclusions that can be drawn from them by the usual thermodynamic processes. And it can be proved conclusively, both from Mr. Dodge's data and in many other ways, that there actually is enough curvature in them to make the contrary assumption wholly misleading."

Dr. Sanford A. Moss Mr. Dodge's experiments, as well as the throttling experiments of Grindley, Griessmann and Peake are exactly the same as the "Porous Plug" experiments made by physicists. In any such experiment there is throttled expansion, that is, expansion from one pressure to another without doing any work and without radiation or final velocity, the "total heat" being constant. Under such circumstances, the ideal "perfect gas" would remain at constant temperature. Actual gases under most circumstances cool when they so expand, however, and experiments of the type under consideration give the amount of this cooling.

2 I have made considerable study of the law connecting this matter with specific heat, which forms the basis of the present paper. The following statement of the law is equivalent to that given in the paper:

3 Suppose we have a gas at a given pressure P'' and a given temperature A'', and suppose we have throttled expansion to another pressure P', the final temperature being A'. Next suppose we make a small addition of heat at constant pressure to the substance when at the original condition so as to give a small temperature increment $B'' - A''$ at pressure P''. Suppose from this latter condition we again have throttled expansion to the previous pressure P', the difference between the final temperatures being $B' - A'$.

4 Since the amounts of total heat were constant during the two throttled expansions, the amount of heat which gave the temperature increment $B'' - A''$ at temperature P' is exactly the same as the amount of heat which would give the temperature increment $B' - A'$

at pressure P'. Hence the ratio of values of specific heat at constant pressure at the two points is the inverse of the ratio of temperature increments. It is, of course, to be noted that the two points whose specific heats are thus related are points having the same total heat, that is, on the same line of throttled expansion.

5 I have plotted a diagram like Fig. 6, with all the points of Table 1, for 15 lb. final pressure. I reduced to exactly the same initial pressure all points with neighboring values of initial pressure, by a formula given later. I find that each of the lines for a given initial pressure is a straight line as nearly as can be judged from the rather irregular points. That is to say, when points from all tests with neighboring values of initial pressure are plotted together as in Fig. 6, a straight line well represents them, instead of a curved line as drawn for the single tests in Fig. 6. From these straight lines I have deduced the empirical laws mentioned later. These laws are all based on the results given in Table 1 and may be considered as mathematical expressions well representing the values of this table.

6 As explained later, the conclusion that the lines of Fig. 6 are straight means that the ratio of the two values of specific heat at the initial and final pressures is constant regardless of the temperatures.

7 The plotted points are so irregular that I question the advisability of selecting specific pairs of points and finding variable ratios for them, as Mr. Dodge does in the last two columns of Table 1. It may be that the lines of Fig. 6 are not exactly straight lines. However, I believe a definite conclusion one way or the other cannot be drawn from the experiments given, owing to their irregularity. Hence straight lines, since they represent the points as well as curved ones, are preferable for simplicity.

8 Since straight lines represent the points, it follows that increments of temperature due to the addition of the same amount of heat at constant pressure at two given pressures have the same ratio, regardless of the temperatures involved. That is, values of specific heat at two given pressures have the same ratio regardless of the temperatures, the proviso being added, however, that the two temperatures have a relation to each other in that they are produced by throttled expansion between the two given pressures. In other words, the ratio of values of specific heats at any two given pressures is constant for any pair of temperatures such that the total heat has the same value. Hence, if lines corresponding to constant values of total heat of superheated steam be drawn on a curve giving values of C_p against temperatures for given pressures, then the ratio of values of

C_p for any one pair of pressures will be the same for each pair of points on a constant total heat line.

9 If the results of throttling experiments are plotted as in the author's Fig. 1, the straight line law means that intercepts between any two lines of "constant total heat" on a pair of constant pressure lines, bear the same ratio to each other. That is, referring to Fig. 1,

$$\frac{A'' B''}{A' B'} = \frac{A'' C''}{A' C'}$$

This ratio is, as already discussed, the inverse ratio of values of C_p for the two pressures (at temperatures corresponding to the same total heat). This is also the ratio of the slopes of the straight lines in the figure referred to similar to Fig. 6. It turns out the actual slopes have very closely the ratio,

$$\frac{C_{p2}}{C_{p1}} = \frac{1460 + P_2}{1460 + P_1}$$

The irregularity of the points makes it impossible to say that they give this law exactly. However, they are closely represented by it. It will be interesting to apply this law to other throttling experiments.

10 If the straight lines mentioned are continued by extrapolation, it turns out that they all intersect at 877 deg. fahr. for both ordinate and abscissa. If this is valid, 877 deg. is the "temperature of inversion" for steam for all pressures. For temperatures above this, there is heating during throttled expansion, such as is well known to occur with hydrogen at ordinary temperatures and which prevented its liquefaction. For temperatures below the temperature of inversion, there is cooling such as occurs with most gases, including hydrogen at low temperatures. Cooling during throttled expansion from high pressures, at temperatures below the temperature of inversion, is the usual means of liquefaction of gases.

11 Of course, this exact value of the temperature of inversion may be incorrect. Even if the lines are practically straight for the region of Fig. 6, they may curve enough to give a higher value.

12 The combination of the two mathematical laws mentioned gives as the relation between pressure and temperature (that is, the equation in Fig. 1) of a line of constant total heat or throttled expansion,

$$(877 - t)\ (1460 + P) = \text{constant} = (877 - t_1)\ (1460 + P_1) \quad (2)$$

Here t is temperature in fahrenheit degrees and P is absolute pressure in pounds per square inch. This relation gives the temperatures at points for which the specific heat ratio (1) holds.

13 Equation (2) is the general law of cooling for porous plug or throttled expansion of steam, and a similar law probably holds with more or less accuracy for other gases. Following are some deductions from this law useful in thermodynamic work or porous plug work such as that of Joule and Kelvin or Rose-Innes.

14 The explicit relation between corresponding drops of pressure in pounds per square inch and temperature in fahrenheit degrees is,

$$t_1 - t = \frac{(P_1 - P)\ (877 - t_1)}{1460 + P}$$

15 The corresponding differential coefficient or variation of temperature with pressure for constant total heat is,

$$\mu = \left(\frac{dt}{dp}\right)_H = \frac{877 - t}{1460 + P}$$

If the temperature is small compared with 877 and if the pressure is small compared with 1460, as is the case when the initial conditions are nearly atmospheric; or if the initial temperature and the final pressure are constant; then the temperature drop is directly proportional to the pressure drop.

16 For all values of temperature, if the pressure is small compared with 1460 or if the final pressure is nearly constant, we have

$$t_1 - t = K\ (P_1 - P)\ (877 - t_1)$$

17 It is to be remarked that in all of the above discussion, the term "total heat" is used for the total energy which has been put into a gas to bring it to a given condition. This includes the internal energy actually present as molecular vibration, etc., as well as the energy expended in passing the gas into the region of the given pressure. If we denote total heat by H and internal energy by U and if A is the reciprocal of the mechanical equivalent of heat,

$$H - H_0 = U - U_0 + A\,P\,V - A\,P_0V_0$$

$$C_p = \left(\frac{dH}{dt}\right)_P$$

Total heat as thus defined is constant during porous plug or throttled expansion.

NOTE.—The author preferred not to present a closure.—THE EDITOR.

No. 1212

THE TOTAL HEAT OF SATURATED STEAM

By Dr. Harvey N. Davis[1] Cambridge Mass.

Non-Member

For many years Regnault's classic formula, now 61 years old, which gives as the total heat of saturated steam

$$H = 1091.7 + 0.305\ (t - 32)\ \text{B.t.u.}$$

has been exclusively used by engineers in this country. This is a well deserved tribute to his extraordinary genius and painstaking skill—no one can possibly read his original memoirs without a new and constantly increasing respect and admiration for him; nevertheless, the remark is becoming common in the literature of engineering research that such and such a method cannot be used because of the well known errors in the steam table. It is therefore fortunate that, at least in the range from 32 to 212 deg., physicists have recently provided a considerable number of good observations of the total heat of saturated steam apparently not noticed by the makers of our steam tables. It is equally unfortunate that, in all these years, there seems to have been not a single new observation above the boiling point. It is the purpose of this paper to show that certain observations recently made for very different purposes can be combined to give a better set of values of H above 212 deg. than do Regnault's direct measurements, and to propose a new formula for the range from 212 to 400 deg., the accuracy of which is believed to be something like 0.1 per cent. If these results are correct, Regnault's formula is too high by more than 18 B.t.u., or 1.7 per cent at 32 deg.; too low by 6 B.t.u., or 0.5 per cent at 275 deg.; and too high again above 380 deg., the error increasing rapidly at high temperatures.

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Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

2 Some years ago, attempts were made to determine the variation of the specific heat of superheated steam with pressure and temperature by means of throttling or wire-drawing experiments. These attempts failed because, as the observers themselves pointed out, the necessary computations were extremely sensitive to small errors in the assumed values of the total heat of saturated steam. Under unfavorable circumstances, an error of 0.1 per cent in one of the values in the steam tables might make a difference of from 3 to 5 per cent in C_p. It is, then, evident that, knowing C_p independently, one could reverse the process by which they tried to get it, and compute all the total heats in terms of any one by a method as insensitive to errors in the assumed data as the other was sensitive. This process will be described later.

3 Fortunately, since these experiments, C_p has been determined independently and directly by Knoblauch and Jakob of Munich and by Thomas of Cornell, and it will presently be shown that the accuracy attained by them is sufficient to make worth while such a recomputation of the wire-drawing experiments.

4 In this work Knoblauch's values of C_p will be used rather than Thomas.' Some reasons for this have been presented to the Society in connection with Heck's recent paper on The Thermal Properties of Superheated Steam, and need not be repeated. Any not persuaded of the wisdom of that opinion may still have considerable confidence in this work, not only because of the general insensitiveness of the method to errors in C_p, but particularly because practically all the values of C_p involved are at low pressures and very moderate superheats, and in this corner of the diagram there is less question as to the accuracy of Knoblauch's values than elsewhere; furthermore because, even had Heck's values been used as printed, the resulting total heat of saturated steam would have been increased by less than 3 B.t.u. at 400 deg., the increase becoming continually smaller as one approached 212 deg. This is not an estimate of the limit of error of the new formula except for those who still believe in the accuracy of Thomas' work near the saturation line.

5 The throttling experiments used are those of Grindley, in England, in 1900, of Griessmann, in Germany, in 1904, and of Peake, in England, in 1905. The agreement which will be shown to exist between the results obtained from these three different sources will be another reassuring proof of the insensitiveness of the method to errors of observation in its assumed data, for the three sets of throttling curves show the obvious differences to be expected in independ-

ent researches. The agreement of results is the more noteworthy when it is remembered that the precautions against radiation and other losses were very different in the three cases. Grindley used a thick glass throttling plate and an independent and independently heated steam "cosie" or jacket, and calibrated his thermometers indirectly by means of Regnault's pressure temperature curve;[1] Peake used a thin mica throttling plate, put wire gauze in the path of the steam to insure thorough mixing, jacketed the low pressure chamber with the escaping steam itself, and used thermometers calibrated by comparison with a previously standardized platinum resistance thermometer; while Griessmann replaced both throttling plate and jacket by a porous plug in a box-wood mount, heavily lagged, in imitation of Joule and Thomson's famous plug experiment, and used thermometers calibrated both by comparison with Reichsanstalt standards and indirectly by means of a steam table.

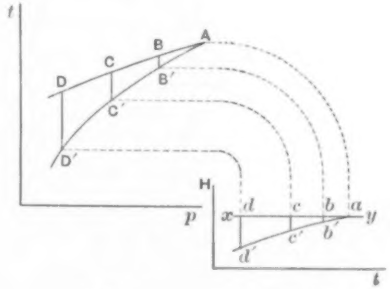

Fig. 1 Fig. 2

Diagram Showing how the Total Heat Curve $yb'c'd'$ is Obtained from a Throttling Curve $ABCD$

6 The method of recomputing these throttling experiments is illustrated by Fig. 1 and 2. Fig. 1 represents a throttling curve of the sort published in the three papers to be discussed. Supposedly dry and saturated steam at the pressure and temperature corresponding to the point A is first throttled to a lower pressure and temperature corresponding to the point B; then in a later experiment in the same run, it is throttled from exactly the same initial condition A to the condition C; then to D and so on. The well known law of throttling is that the total heat in the condition B, or C, or D, is equal to that in the initial condition A.

[1] As an extra precaution, all of his temperatures have been recomputed for this paper with the best modern data.

7 The point B represents superheated steam at the pressure p_B; the point B' represents saturated steam at the same pressure; and the amount of superheat at B is the known temperature there minus the temperature at B', which can be taken from a steam table. Also, by definition, the total heat at B equals that of saturated steam at the same pressure (point B') plus the amount of heat required to superheat it at constant pressure from B' to B. This is the integral of C_p from B' to B, or simply the mean C_p from saturation multiplied by the known superheat. If C_p is known, this integral, or increment in the total heat between B' and B, is easily evaluated.

8 This integral is not only the difference between the total heat of saturated steam at B' and that of superheated steam at B; it is also the difference between the total heat of saturated steam at B' and that of saturated steam at A; that is, between the two corresponding ordinates of the curve that gives the total heat of saturated steam as a function of the temperature, the curve sought in this paper. To draw a piece of this curve. one chooses arbitrarily some horizontal line such as xy in Fig. 2, and lays off below it, at the proper temperatures, the distances bb', cc', dd', etc., which represent on the desired H-scale the integrals or total heat differences between B' and B, C' and C, D' and D, etc. Th curve $ab'c'd'$ is an isolated piece of the true curve of total heat against temperature. The *relative* height of its points, that is, its shape, is accurately determined; the *absolute* height above the usual zero of total heats, namely, that of water at 32 deg. fahr., is as yet wholly unknown. The experiments of Grindley gave seven independent sample pieces of this sort, one for each throttling curve, their temperature ranges being known and greatly overlapping; similarly Griessmann's data gave eleven such sample pieces, and Peake's six.

9 One chief difficulty in the original work was that the steam at A was often not quite dry, so that its condition was not determined by the observed pressure and temperature. All the initial points, such as a, have therefore been omitted from the sample curves. This does not affect the validity of the reasoning about the remaining points, as $b'c'd'$, etc.

10 All sample pieces of any one observer were then plotted carefully on very thin transparent rice paper, with vertical guide-lines at certain standard temperatures, which enabled these plots to be accurately oriented as far as rotation and horizontal displacement were concerned, but left them free to slide up and down over each other. The sheets were then piled on top of one another on a trans-

parent table lighted from below, each one placed so as to make its piece of curve coincide most satisfactorily with the overlapping pieces already laid down. The exact relative displacements of the sheets were then carefully measured. This process was repeated for each of the three observers' sets of sheets independently, four different times for each set, in two very different orders and in those orders reversed on different days, all with the object of avoiding as far as possible any routinizing effects of memory or habit which might disturb the real independence of the four determinations. The means of the measured displacements were then used to reduce each of the pieces of curve in any one of the sets arithmetically, not graphically, to a zero common to all the curves of that set. The results are marked *Gy*, *Gs*, and *P* in Fig. 3. They are plotted separately for clearness, but they are simply different experimental determinations of exactly the same real curve. The vertical scale of each is four calories[1] to the square. The height of each above its true zero is still unknown. Each of the circles represents at least one independent throttling observation, and some few of them two or three independent observations that happened to coincide. It will be noticed that no one of the curves is more than a tenth of a square wide between centers. Each therefore seems a good determination of the true curve within two-tenths of a calorie, which is less than 0.4 B.t.u.

11 The next step was to establish a comparison between the three curves. The points of each were first grouped in segments of some 20 deg. and the mean point of the group was used to represent the group. There were eighteen such means, seven representing Grindley's points, five Griessmann's and six Peake's. These means were then plotted to scale on three sheets of rice paper and fitted together in the way already described. The result is shown in the next lower curve in Fig. 3. The absolute height of the curve as a whole is still arbitrary. The reader can now judge for himself as to the agreement of the results obtained from the three independent sources.

12 The formula chosen to represent this curve is of the second degree in the temperature. This form is justified and was indeed suggested, by plotting on a single diagram the derivatives of the three curves *Gy*, *Gs* and *P* considered separately. These derivatives agree remarkably with each other in determining a straight line, much

[1] For convenience all diagrams are left in the metric units in which they were computed.

better than could have been expected of the derivative of an empirical function. A more complicated representation is wholly unnecessary, and unwarranted by their limit of error.

13 Assuming the general formula

$$H = H_{212} + a\,(t - 212) - b\,(t - 212)^2$$

the constants a and b were determined by the method of least squares, using all eighteen mean points reduced to a common but arbitrary zero. The results were checked by the usual least squares methods, practically insuring accuracy; but as a rough preliminary check the constants were determined from three random points, one at each end of Grindley's curve and one in the middle, and these coefficients agreed with those afterward obtained within a third of one per cent of their own values, which means less than one hundredth of one per cent in H itself throughout the range of the formula. The agreement of the three curves with their mean is obvious.

14 The resulting formula, in English units, is

$$H = H_{212} + 0.3745\,(t - 212) - 0.000550\,(t - 212)^2$$

This formula gives the total heat of saturated steam between 212 deg. and about 400 deg. *in terms of that at 212 deg.* A value for this fundamental constant H_{212} will presently be chosen from those available in the literature of the subject, but it should be remembered that even if this choice is wrong or if new and different data near 212 deg. are hereafter published, whatever merit the above equation may have will be wholly unaffected by the necessary change in H_{212}.

15 It is interesting to compare the self-consistency of this work, as represented by the narrowness of the bands of plotted points, with that of Regnault's observations, which are plotted at the bottom of Fig. 3.[1] His band is at least eight or ten times as wide as any of those above it. It should also be noticed that something evidently happened to his apparatus at 178 deg. cent. and that allowing for this, his band shows unmistakably the same curvature as those above it. The observations above 178 deg. cent. were, as a matter of fact, the last he made, and he speaks definitely of serious trouble with his apparatus at the very point at which the jump occurs; in fact he had to renew many of its parts, and to watch it continually thereafter, so that his conditions may well have been somewhat

[1] The large circle at the boiling point, 100 deg. cent., represents the mean of 38 points, of which only the highest and lowest are plotted.

changed. This discontinuity in his curve has been noticed by many writers, one of whom attributes it to a leak in his distributing valve, remedied at this point, but this is not definitely mentioned in the memoir.

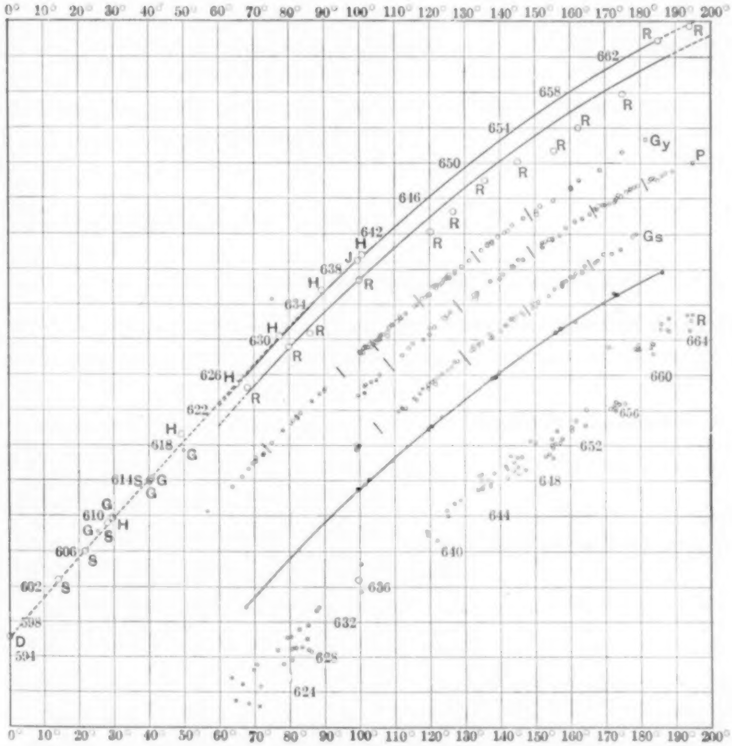

Fig. 3 Curves of the Variation of Total Heat with Temperature (in Calories against Deg. Cent.)

GY, P AND GS ARE CURVES FROM THE DATA OF GRINDLEY, PEAKE AND GRIESSMAN. THE LINE BELOW THEM SHOWS THEM SUPERPOSED. AT THE BOTTOM (CURVE R) ARE REGNAULT'S OBSERVATIONS PLOTTED ON THE SAME SCALE FOR COMPARISON. AT THE TOP ARE THE VALUES OF DIETERICI (D), A. W. SMITH (S), GRIFFITHS (G), HENNING (H), JOLY (J), AND REGNAULT (R), WITH THE CURVE BELOW REPRESENTED AS ATTACHED BOTH TO REGNAULT'S VALUE AT 100 DEG. CENT. AND TO THE MEAN OF HENNING'S AND JOLY'S VALUES. THE LATTER IS SEEN TO BE PREFERABLE

16 All results that could help in the evaluation of the fundamental constant H_{212} have recently been admirably summarized by Professor Smith of the University of Michigan, whose paper has been most useful throughout this work. These results are collected in

Table 1 and are plotted as the top curve in Fig. 3, with the movable section that has been built up in this paper represented as attached, first to Regnault's mean at the boiling point, and second to the mean of the values of Henning and Joly. It will be noticed, first, that in either case most of Regnault's values are too low;[1] this was to be expected, as we have begun only recently to realize the difficulty in removing the last traces of unevaporated water from apparently dry

TABLE 1 SUMMARY OF RESULTS FOR EVALUATION OF FUNDAMENTAL CONSTANT H_{212}

Observer	Temperature degrees fahr.	Total heat B.t.u.
C. Dieterici, Königliche Technische Hochschule, Hanover, Germany	32.0	1073.4
A. W. Smith, University of Michigan, U. S. A	57.1	1084.7
	70.1	1090.7
	82.5	1096.2
	103.6	1104.6
E. H. Griffiths, Cambridge, England	86.0	1097.8
	104.3	1104.9
	76.9	1091.5*
	103.9	1104.2*
	121.7	1111.2*
F. Henning, Reichsanstalt, Berlin, Germany	86.2	1097.6*
	120.5	1114.4
	148.7	1124.7
	171.2	1134.5
	192.7	1144.0
	213.1	1151.1
J. Joly, Trinity College, Dublin, Ireland	211.9	1150.0

*These four values were considered by the experimenters less reliable than their other results.

steam, and any such moisture would make the results too low; it is therefore probable that his results at the boiling point are also too low for the same reason; second, that if Regnault's mean at 100 deg. cent. is right, his values at the highest temperatures are too high, a deviation hard to explain on any ground.[2] This makes it still more probable that the true value at 100 deg. is at least as high as the mean of the values of Henning and Joly; and third, that although the

[1] They are represented, in groups, by means.

[2] It is hard enough to see why they should suddenly begin to run as high as on the curve; this raises the only doubts that have yet appeared as to the wisdom of the choice of the C_p values used.

less accurately determined lower end of the movable section does not fit particularly well among the fixed points in either case, the agreement is much more satisfactory when the section is oriented on the mean of Henning and Joly. Finally, both of these observers are among the most skillful experimenters of their time, as Regnault was in his; both have had the advantage of improvements in the technique of accurate measurements accomplished in the several decades since Regnault; and both have used methods differing from each other's and from his, and superior to his in sensitiveness and accuracy. For all of these reasons, the mean of their values, namely 1150.3 B.t.u., seems the most probable value of the total heat of saturated steam at 212 deg. fahr.

17 The final formula, in English units, is

$$H = 1150.3 + 0.3745\,(t - 212) - 0.000550\,(t - 212)^2$$

Its range is from 212 to about 400 deg. fahr.; it does not pretend to accuracy below 212 deg.; in that range an accurately drawn graph

TABLE 2 NEW FORMULA COMPARED WITH STANDARD VALUES

Temperature Degrees fahr.	Pressure pounds	Total Heat		Specific Volume	
		New	Peabody	New	Peabody
32.	0.089	1073.4	1091.7	3294.	3395.
100	0.949	1103.6	1112.4	349.7	354.7
212	14.70	1150.3	1146.6	26.76	26.66
281.0	50.	1173.6	1167.6	8.496	8.429
327.8	100.	1186.3	1181.9	4.412	4.409
358.5	150.	1193.4	1191.9	2.998	3.016
381.9	200.	1198.1	1198.4	2.277	2.299
401.0	250.	1201.5	1204.2	1.837	1.858
417.4	300.	1204.1	1209.3	1.540	1.558
426.3	330.	1205.3	1211.9	1.403	1.417

should be used. The dotted line in Fig. 3 indicates the position of what seems the most probable graph; in drawing it Regnault's observations had to be ignored altogether; the agreement among the other more recent authorities is then most satisfactory.

18 The new formula, supplemented by such a graph, is compared with the standard values given, for instance, by Peabody or Reeve, in Table 2 and Fig. 4 and 5. The table gives the new and the old values of the total heat at various temperatures from 32 to 425 deg.,

and also the new and the old values of the specific volume of saturated steam as computed by the usual Clapeyron equation

$$V - v = J\frac{r}{T}\frac{1}{\frac{dp}{dt}}$$

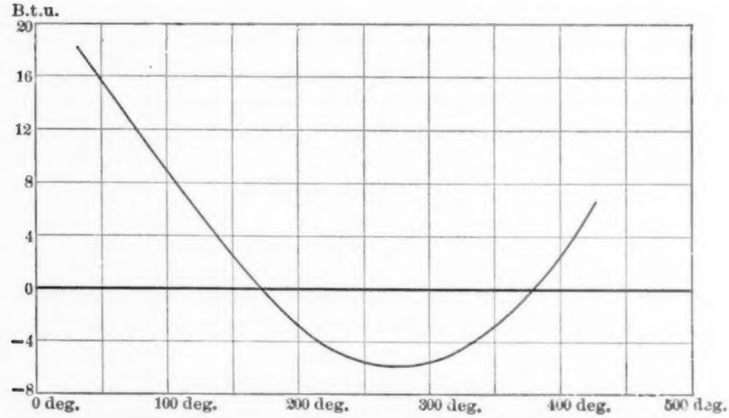

FIG. 4 DEVIATION FROM THE AUTHOR'S VALUES, OF REGNAULT'S VALUES OF TOTAL HEAT OF SATURATED STEAM, GIVEN IN PEABODY'S TABLES. HIGH TEMPERATURE END OF CURVE IS A PARABOLA

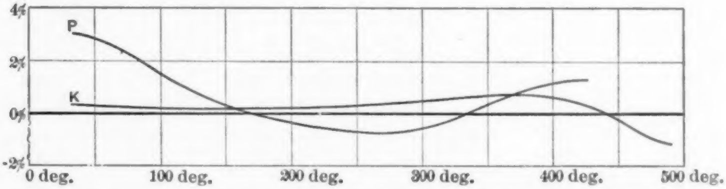

FIG. 5 PERCENTAGE DEVIATION OF REGNAULT'S VALUES OF SPECIFIC VOLUME OF SATURATED STEAM, GIVEN IN PEABODY'S TABLES, AND OF KNOBLAUCH'S EXPERIMENTAL VALUES, FROM THE NEW VALUES—EVERYTHING ABOVE 400 DEG. EXTRAPOLATED

The necessary values of dp/dt were taken from Henning's recent and valuable review of the available literature.[1] In Fig. 4, the deviation of the standard values of H from those here presented is shown graphically. In Fig. 5 the percentage deviation of the standard values of V from the new ones is similary presented. This diagram also shows, in the curve marked K, the deviation of Knoblauch's experimental

[1] Wied. Ann. 22 (1907), p. 609.

values, as given by Linde's equation,[1] from those of this paper. The agreement is gratifyingly good up to 400 deg.; above that both equations are extrapolated, so that some disagreement is not surprising.

19 These total heat values can be made to throw new light on a subject recently discussed before the Society by Professor Heck, namely, the specific heat of superheated steam. It has several times been pointed out in the literature that, *along the saturation line,*

$$C_p = \frac{dH}{dt} - \frac{r}{T} + \frac{r}{u}\left(\frac{dv}{dt}\right)_p$$

where H is the total heat of saturated steam, r its heat of vaporization, u the change of volume during vaporization, t the ordinary and T the absolute temperature, and $\left(\frac{dv}{dt}\right)_p$ a derivative out into the superheated region like C_p itself. Not much use has hitherto been

TABLE 3 VALUES OF C_p PLOTTED IN FIG. 6

Pressure pounds	Temperature degrees	Cp (calculated)	Cp (Knoblauch)	Cp (Heck)
15	213	0.484	0.470	0.581
35	259	0.506	0.492	0.614
75	308	0.560	0.537	0.650
115	338	0.615	0.584	0.680
195	380	0.722	0.699	0.719
250	401	0.794	0.795	0.739
310	420	0.875	0.919	0.758
370	437	0.956	1.068	0.776
450	457	1.067	1.318	0.798
530	473	1.179	1.634	0.819
610	488	1.293	2.041	0.842

made of this equation because of the outstanding errors in Regnault's values for saturated steam, but it is hoped that the new values of H, r, and u of this paper are good enough to make the computation worth while. The volume derivative is a source of some uncertainty, but fortunately its value needs to be known only on the saturation line itself, and there Linde's characteristic equation for superheated steam is at its best, and seems satisfactory. Certain values of C_p, computed in this way, are given in Table 3 and are plotted with circles in Fig. 6.

20 This figure also shows the saturation curves of Knoblauch, of Thomas and of Heck. Knoblauch's curve is extrapolated analyt-

[1] See the Forscharb, Berlin, Heft 21, p. 67; or Peabody's Steam Tables, 1907 ed., p. 21.

ically from four points at low pressures, which are themselves extrapolated from observed points at various favorably situated superheats. The extrapolations to saturation may be uncertain by small amounts which are represented in Fig. 6 by short vertical lines attached to Knoblauch's curve. The extrapolation to very high pressures is of course uncertain.

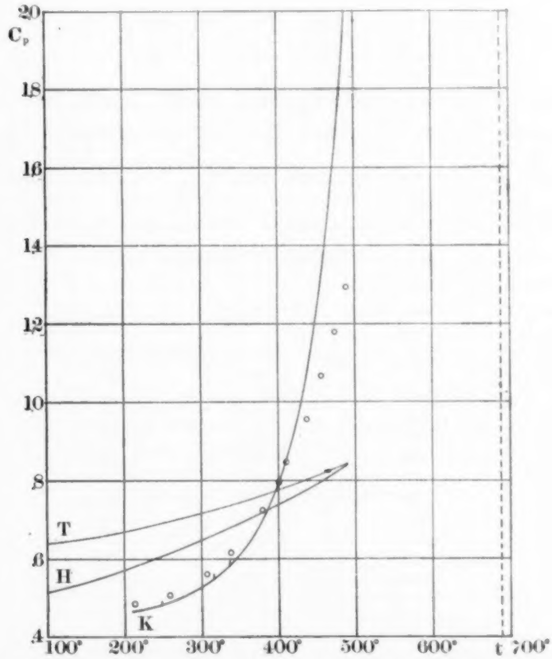

Fig. 6 Representing the Saturation Line on the Regular C_p Diagram, as Drawn by Knoblauch (K), by Thomas (T), and by Heck (H) The Circles Represent Values Computed Thermodynamically. The Dotted Line at the Right, at t = 689 Deg. = The Critical Temperature, Should be Approached as an Asymptote by the True Curve

21 Thomas' curve has even less experimental foundation, if the criticisms already presented in connection with Heck's paper are justified, and would seem to be a sheer guess. Heck's line was presented as a compromise between Knoblauch's and Thomas'. If Thomas' line is discredited, Heck's ceases to be interesting.

22 In view of all this uncertainty the evidence furnished by the thermo-dynamic computation described above would seem to be

of considerable importance. As Fig. 6 shows, the computed points agree with Knoblauch's curve rather than with those of Thomas or Heck, up to about 400 deg. (250 lb.). Above that temperature both Linde's equation and the total heat equation of this paper are extrapolated, and so the computation means very little, but even at high temperatures the general character of the computed curve is more like Knoblauch's than like Thomas' or Heck's.

23 This computation furnishes therefore an additional justification of the use of Knoblauch's values in this paper. It will perhaps be argued that this justification is simply a circular fallacy—in that a set of total heat values based on Knoblauch's values of C_p might be expected to lead back to Knoblauch's values at the end—but a careful consideration of the formulae involved will show that the objection is only in a very small measure well founded, and that the circularity of the reasoning is apparent, not real.

SUMMARY

24 The throttling experiments of Grindley, Griessmann and Peake, considered in connection with Knoblauch's determination of the specific heat of superheated steam, lead to a new formula for the total heat of saturated steam, namely

$$H = H_{212} + 0.3745\,(t - 212) - 0.000550\,(t - 212)^2$$

25 The best available value of H_{212} seems to be 1150.3 mean B.t.u., which is the average of the values of Henning and of Joly. The total heat equation becomes

$$H = 1150.3 + 0.3745\,(t - 212) - 0.000550\,(t - 212)^2$$

The range of this formula is from 212 to about 400 deg. fahr. The greatest error in Regnault's formula in this range is 6 B.t.u., at 275 deg. fahr., but if Regnault's formula is extrapolated to higher temperatures, the error in it increases very rapidly. Below 212 deg. there is an abundance of modern data to show that Regnault's formula runs high, the error reaching 18 B.t.u. at 32 deg.

26 Recomputed values of the specific volume of saturated steam differ from the standard values by 101 cu. ft. or 3 per cent at 32 deg. and by about 1 per cent in the opposite direction at 275 deg. Com puted values of C_p at saturation agree strikingly with Knoblauch's values, and give additional confirmation to a conclusion already pre-

sented to the Society, that of the three available sets of C_p values, Knoblauch's, Thomas' and Heck's, Knoblauch's is most deserving of confidence.

27 A steam table based on these new values will presently be published under the joint authorship of Professor Lionel S. Marks and the present writer.

DISCUSSION

PROF. C. H. PEABODY[1] This paper is so complete and conclusive that it needs no discussion; rather it is to be accepted as the most valuable contribution to the science and practice of steam engineering since the determination of the mechanical equivalent of heat by Rowland.

2 To my mind this piece of work, which cannot be appreciated too highly, emphasizes two features; first, that no good scientific work is ever wasted, and second, that the highest scientific ability is required to interpret and apply experimental data. It would add to the value of the paper if the author would append the references to the authorities quoted, somewhat more fully than he has done.

3 In consequence of the information presented by Dr. Davis it will be necessary to recompute our steam tables; in fact his paper informs us that a new table is to be published, which I am sure will be welcomed by engineers.

4 It may, however, be pointed out that existing tables are in error only to the extent of half of one per cent for the middle range of temperatures, and that such errors will give engineers but little concern, however distasteful they are to the computers of such tables.

5 But our temperature-entropy diagram and the temperature-entropy tables (for which I am responsible) need change in only one feature and that the one of least importance.

6 To show that this is true let us consider the usual expression for entropy of wet steam,

$$\frac{xr}{T} + \theta$$

x = quality or dryness factor.
r = heat of vaporization.
T = absolute temperature.
θ = entropy of the liquid.

[1] Professor of Naval Architecture and Marine Engineering, Mass. Inst. of Technology, Boston, Mass.

7 In computations for a temperature-entropy diagram or table we begin by assigning some desired value to the entropy which remains constant for a given abscissa or column. Then

$$\frac{xr}{T} + \theta = \phi$$

$$\therefore\ xr = (\phi - \theta)\, T$$

which determines the product xr for any temperature and entropy even though the factors x and r should be unknown. Consequently the heat contents

$$xr + q = (\phi - \theta)\, T + q$$

will not be affected by changes in r.

8 Again the usual manner of computing the specific volume of saturated steam is by the equation

$$s = \frac{r}{A\,T}\,\frac{1}{\frac{dp}{dt}} + \sigma = n + \sigma$$

s = specific volume of saturated steam.
A = reciprocal of Joules equivalent.
$\frac{dp}{dt}$ = slope of temperature-pressure curve.
σ = specific volume of water.
u = increase of volume due to vaporization.

9 Now the specific volume of wet steam is

$$v = xu + \sigma = \frac{xr}{A\,T}\,\frac{1}{\frac{dp}{dt}} + \sigma$$

or substituting for xr its value

$$v = \frac{(\phi - \theta)\, T}{A\,T}\,\frac{1}{\frac{dp}{dt}} + \sigma$$

$$\therefore\ v = \frac{\phi - \theta}{A}\,\frac{1}{\frac{dp}{dt}} + \sigma$$

which shows that the specific volume for a given temperature and entropy will not be changed by a change in r.

10 On the contrary the quality or dryness factor

$$x = (\phi - \theta) \frac{T}{r}$$

depends directly on r.

11 These equations are deduced to show that of the several properties given on a temperature-entropy diagram or table, only one, namely the quality, needs revision. The fact that the initial value of this factor is seldom known to the degree of certainty represented by half of one per cent has no particular bearing on this discussion unless it makes engineers somewhat impatient concerning it.

Prof. William D. Ennis Professor Peabody has said what was fitting, and what he could most appropriately say, regarding this masterly paper. When the aims and methods of pure science can be as helpfully presented to engineers as Dr. Davis has presented them, we must derive inspiration. Two questions immediately arise in reviewing these revised values for the total heat of steam. First, are they of considerable engineering importance? Second, are the new values final? On the first point: Even if we take H_{212} at 1150.3 the difference between the new and old values within ordinary ranges is small. Take, for example, an engine test showing a thermal efficiency of 0.1500, using saturated steam at 150 lb. pressure. The old and the new values of H are respectively 1191.9 and 1193.4; a difference which would make the thermal efficiency, based on the new value, read 0.1498. At the same time, the new values differ from the old to such an extent as to promise some noticeable variations in our steam tables.

2 As to the finality of Dr. Davis' deductions, it seems unquestioned that throttling methods for the determination of H are better than the older method, *provided* the values of C_p are accurately known and there is no question as to the relation between p and t at saturation. But are the values of C_p as yet established? Professor Heck has harmonized the two best sets of experiments and has regarded the question as "about settled." Dr. Davis also regards the question as settled, but in a different way; while Professor Thomas evidently holds it to be unsettled, because he is still experimenting. We cannot get final values of H until we have final values of C_p.

3 I am not quite clear as to whether the apparent check on

Knoblauch's values of C_p is not after all in large measure an example of the circular fallacy. The analytical expression for C_p includes three terms. The first of these, $\frac{dH}{dt}$, may be taken from Regnault's formula or from the new formula; the value in the latter case depends quite directly upon C_p. The second term of the expression for C_p, $\frac{r}{T}$, also depends directly upon C_p, for $r = H - h$ and H has been computed from C_p. For the same reason, the third and last term also depends upon C_p, although the derivative $\left(\frac{dv}{dt}\right)_p$ may be obtained without regard to Knoblauch's values for C_p. The computed values of C_p thus depend, though not simply, upon the values assumed for C_p in the first place. We could, of course, obtain a great variety of curves like that suggested by the small circles in Fig. 6, according to the origin of our values of H and $\left(\frac{dv}{dt}\right)_p$. I have found, for example, using Regnault's values for H, p, t and Wood's formula of relation between p, v and t for the derivative, at 140 lb. absolute pressure, that $C_p = 0.622$; a value rather closer to Knoblauch's than Dr. Davis' computation would give. This strikes one as being purely accidental.

4 With correct values of C_p, there seems to be no possible question as to the accuracy of re-computing H by the proposed method. The best check on the whole work would be, then, to finally determine H directly by some appropriate method.

Prof. Robert C. H. Heck I have not been able to give this subject the amount of consideration which it ought to have, as a preliminary to close quantitative criticism; but several points have occurred to me as worthy of general remark.

2 Dr. Davis, having made a close study of the data along this line, is highly competent to express an opinion as to how nearly the data which he has used are to be accepted as final. We may well accept his conclusion that further changes in the determined values of the specific heat of superheated steam will not produce any great changes in his total heat. Here the word "great" is used from the view-point of scientific precision, not in the engineering sense; and from this point of view the errors in Regnault's formula are very great. But whether the range of probable error, or of uncertainty, is as narrow as Dr. Davis thinks, is to my mind rather doubtful.

3 One fact that has been brought out in all the more useful experiments, especially those of Knoblauch and Jakob, is the extreme difficulty of actual physical realization of that state of steam, so simple in idea, known as dry saturation. This shows up very clearly when Regnault's values are plotted for comparison (Fig. 3 of the paper); consistently, his results fall below the others, indicating the probability, which has frequently been remarked, that his steam was not really dry.

4 Knoblauch and Jakob put their steam through a preliminary superheater, of the form of a long vertical cylinder, with a succession of "pine-tree" radiators, made of glass tubes on metal frames, and with the electrical conductor coiled on the glass tubes; current could be passed through as many as desired of these radiators, and the rest left dead; and the temperature of the current of steam could be measured at the dividing plane between each pair of radiators. It was found that the steam rose in temperature in passing the successive active heaters, but in the dead range it at first dropped off a little and then settled to uniformity from point to point along the line of flow. The drop after leaving the region of heat-impartation shows that sensible heat was being taken up in the steam-current, as by evaporation; and was explained as indicating the presence of moisture or of saturated steam in the body of quite highly superheated steam, until sufficient time, with thorough mechanical mixture, had produced homogeneity.

5 In the experiments of Professor Thomas, the steam was passed through a number of small holes (in effect, tubes), where heat was imparted to it by electrical heater-coils. In one experiment the steam was brought just to the point where any more heat would make the temperature begin to rise above that of saturation; in the next, the steam was heated to some higher point; and the difference in energy consumed was the heat for superheating from saturation. Aside from any question as to accuracy in observation and in measuring instruments, it is legitimate to be doubtful, first, whether the steam is homogeneously dry-saturated in Experiment A; second, whether it is homogeneously superheated in Experiment B.

6 In the throttling calorimeter, the steam at first flows through the orifice in practically adiabatic expansion, some of it being condensed in the operation; then, as the jet comes to rest in the low-pressure chamber, the kinetic energy gained in that first operation is changed back to heat, and the body of steam is thereby superheated.

7 Now the important question is, may we safely assume that the steam in the low-pressure section of the throttling calorimeter is homogeneous when its temperature is measured? The best case for the defense is made when a porous plug is used instead of an orifice, as in the experiments of Griessmann. In general, though, the probabilities appear to be more against the throttling method than against that of Thomas. Under the excellent work which Dr. Davis has done lies this uncertainty as to the inherent reliability of his data.

8 I turn now to the question of the specific heat of superheated steam near the saturation limit—assuming that this limit exists as a sharply defined line, and can be experimentally realized. In the paper which I presented at the Detroit meeting, an attempt was made to combine the results of the best experiments to date. The most uncertain thing about the operation of superheating was the starting point; but I had to have something to start from, and so what seemed the best and most intelligent guess was made. It was, to a considerable degree, however, just a guess, although with the redeeming feature that the resulting uncertainty was much less than the probable error in the total heat up to saturation. In the condition of the data, the best that could be aimed at was essential correctness for engineering purposes, with a judicious balancing of indications and probabilities. The present paper steps upon a higher plane; and with its results fully confirmed, we shall be ready to go out into the region of superheat and really "get things down fine."

9 There is one idea to which I must again take exception, and this is the assumption that the initial specific heat of superheated steam under constant pressure must rise to infinity at the so-called critical temperature, 689 deg. fahr. Infinite specific heat is the characteristic of the ordinary mixture of steam and water, because such a mixture can absorb heat at constant pressure without rise of temperature. This property disappears, however, at the beginning of the critical state; and when it has disappeared from its proper *habitat,* to import it into the foreign region across the boundary appears to be rather unjustifiable.

10 One point further is to be noted, which even yet is, however, of little more than theoretical interest. The total heat which remains constant in a throttling operation is not quite the same as that which was measured by Regnault in his calorimeter and which is given in our steam tables. In the throttling calorimeter, what we may call the work of the feed-pump is included in the total heat. This total

heat, which remains constant in the ideal case of no-radiation, comprises not only the internal or intrinsic energy, but also the external energy of expansion under constant pressure, measured by the product of pressure by volume. In Regnault's experiments, steam was generated in a little boiler, and passed at once into a calorimeter, where it was condensed and cooled, the whole operation taking place under full pressure· then the heat gotten out of the steam and measured was, in intent, just what is put into the steam in the ordinary boiler, but did not include the work done by the feed-pump in forcing the water into the boiler. Until our experimental data are much more reliable than any now available, this small difference remains of theoretical rather than practical importance: but it enters into every precise expression for the energy of the steam-jet, and must be taken into account in calculations.

Prof. Lionel S. Marks For a long time it has been evident that Regnault's values for the total heat of saturated steam require some revision. Particularly is this true for steam of low pressure. Forty years ago Herwig[1] pointed out that the values of the total heat below 120 deg. fahr. were all too low. In his low pressure experiments, Regnault's method of measuring the temperature of the evaporating water by the vapor pressure in the condenser, has very properly given rise to criticism. At higher temperatures the break in the experimental results is clear evidence of the existence of some notable error.

2 But it is not only on the score of inaccuracies in his determinations that Regnault's work has been subjected to adverse criticism. Many of the students of his work who have accepted as correct his experimental results, have found themselves unable to accept his interpretation of his results by a straight line law connecting total heats and temperatures. If the observations above 178 deg. cent. are set aside (on account of trouble with the apparatus at that temperature) it will be seen from Fig. 3 of the paper that Regnault's points do not lie on a straight line, but on a curve which resembles closely the Davis curve. Several physicists in recent years have found that a second degree equation gives the best representation of the relation of total heats and temperatures found by Regnault.

Thus Wüllner[2] proposes

[1] Herwig Pogg. Ann. Vol. 137. 19, 592. 1869.

[2] Wüllner Lehrbuch der Experimentalphysik. Vol. 2, 773. 1896.

$$\lambda = 589 \quad + \; 0.6003\, t - \; 0.001246\, t^2$$

Ekholm[1] gives

$$\lambda = 596.75 + 0.4401\, t - 0.000634\, t^2$$

and Starkweather[2] finds from Regnault's observations

$$\lambda = 603.2 \quad + 0.356\, t \quad - 0.00021\, t^2$$

for temperatures above 100 deg. cent., and

$$\lambda = 598.9 \quad + 0.442\, t \quad - 0.00064\, t^2$$

for temperatures below 100 deg. cent. (These equations are for 15 deg. calories and centigrade degrees.)

3 In his investigations of other liquids Regnault gave second degree equations for the relation between total heat and temperature in almost every case. It was the results of his trouble with his apparatus at 178 deg. cent. that forced him to give a straight line relation between the total heat and temperature of saturated steam. It will be seen that the Davis equation representing the relation between the total heat of steam and its temperature between 212 deg. and 400 deg. fahr. and based upon the work of a number of modern investigators is of the same form as those given by the most recent analysis of Regnault's work.

4 Other equations have been proposed in recent years based entirely or partly upon other investigations than those of Regnault. Dieterici[3] has deduced an equation for r based on his own experimental value at 0 deg. cent., on Regnault's work, and on certain theoretical and empirical conclusions. The equation is,

$$r = 5948 - 0.559t - 0.000\,002\,234t^3$$

and it shows the same kind of deviation from Regnault's straight line law as the other cited equations. One of the most prominent is that of Thiesen[4] which gives an expression for the value of the latent heat of vaporization of any liquid in terms of the critical temperature, T_c,

$$r = k\,(T_c - T)^{\frac{1}{3}}$$

[1] Ekholm, Fortschritte der Physik. Vol. 46 II, p. 371.

[2] Starkweather, Am. Jour. of Science (4) Vol. 7, p. 13. 1899.

[3] Dieterici, Z.V.D.I., 49 (1905), p. 362–7.

[4] Thiesen, Verh. des Phys. Ges. zu Berlin, 1897–8.

for water,

$$\log_{10} k = 1.924, \text{ and } T_c = 638 \text{ deg. cent.}$$

Besides this Linde[1] has deduced the values of the latent heat of vaporization from the specific volume determinations of Knoblauch, Linde and Klebe, by the Clapeyron equation

$$r = A\,T.\frac{dp}{dt}\,.\,u$$

The relation between the values of r, as found by Regnault, by Thiesen, by Linde and by Davis are shown in Fig. 1. (Below 212

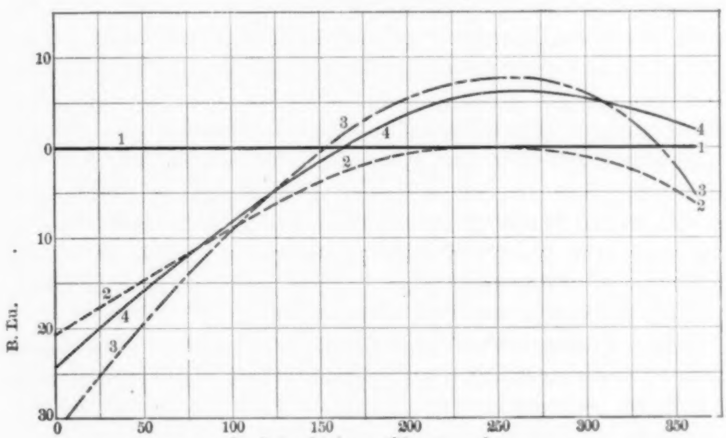

FIG. 1 VARIATIONS FROM REGNAULT'S VALUES OF r; CURVE 1–1, REGNAULT, 2–2, THIESEN; 3–3, LINDE; 4–4, DAVIS.

deg. fahr. the Linde curve is continued through the experimental points of Griffiths and Dieterici.) It will be seen that the Thiesen, Linde and Davis curves show deviations from Regnault's values which have the same general characteristics.

5 Above 400 deg. fahr. there are no reliable experimental observations. If the Davis formula were assumed to be true for temperatures above 400 deg. fahr. it would lead to a maximum value of the total heat at 552 deg. fahr. The critical temperature is 689 deg. fahr. There is no direct experimental evidence to show that the total heat goes through such a maximum and deductions from characteristic equations cannot be used, as they, of necessity, must be widely extra-

[1] Linde. Mitt. über Forschungsarbeiten, 1905, No. 21, p. 71

polated to give any evidence in that region. The value of the total heat must however reach a maximum before the critical temperature.

6 It can be shown that at the critical temperature $\frac{dH}{dT}$ is negative. The isothermal for the critical temperature is generally assumed to be horizontal (on the *pv* plane) where it meets and is tangent to the steam dome. If that is so, the value of $\frac{dH}{dT}$ is minus infinity at the critical temperature. If the critical isothermal is not tangent to the steam dome where it meets it, it must be because the steam dome is

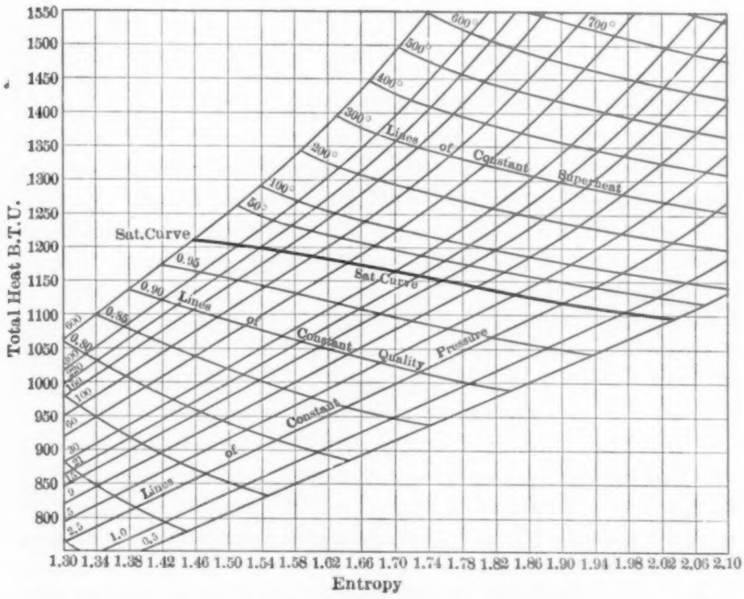

FIG. 2 MOLLIER DIAGRAM SHOWING TOTAL HEAT AND ENTROPY OF STEAM, BASED ON TOTAL HEAT VALUES OF DR. DAVIS

not rounded on top but comes to a peak at the intersection of the water and saturated steam lines. In this case the value of $\frac{dH}{dT}$ at the critical temperature will be finite but still negative. In either case the value of $\frac{dH}{dT}$ is negative and must have gone through zero in approaching the critical temperature, or in other words the total heat, H, must have gone through a maximum. Just where that

maximum value occurs, there is no direct evidence to show either for water or for any other liquid. It is probable that the maximum value does not occur as far from the critical temperature as an extrapolation of the Davis formula would indicate, though there is no experimental evidence to support this opinion.

7 The method that has been used in this paper for finding the variation of the total heat of saturated steam with the temperature is a method capable of giving very accurate results. The remarkable agreement of the results from the three separate sets of throttling

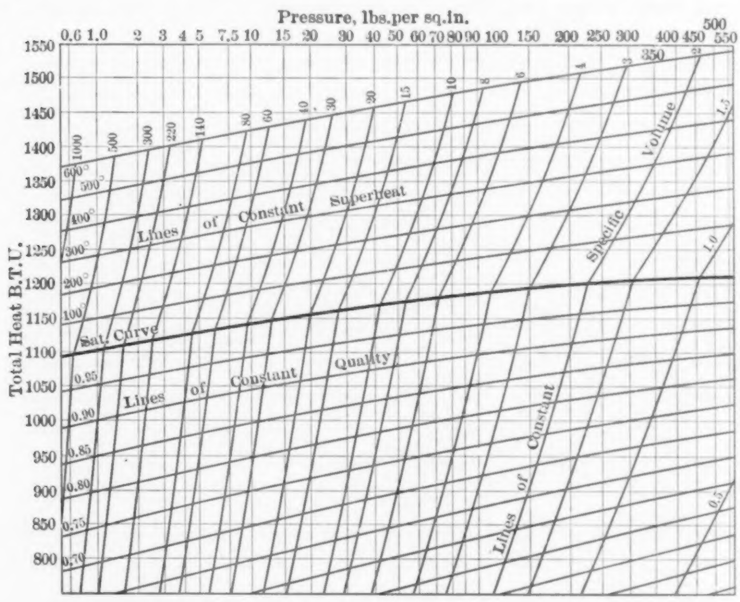

FIG. 3 TOTAL HEAT-PRESSURE DIAGRAM SHOWING SPECIFIC VOLUMES, BASED ON VALUES OF DR. DAVIS

experiments is valuable evidence on that point. The accuracy depends, however, on the use of the proper values for the specific heat of superheated steam. As Dr. Davis has pointed out, there is very little question as to the correctness of the Knoblauch values of specific heat at low pressures and moderate superheats. For higher pressures and moderate superheat, if the Knoblauch values are somewhat high (as the thermodynamically computed values of C_p at the saturation line, author's Fig. 6, would indicate) the effect will be to make the Davis total heat curve (author's Fig. 3) somewhat low at high pressures.

It will be seen that the Davis curve goes close to the two Regnault (R) circles above 180 deg. cent. It is not known exactly what changes were made by Regnault in his apparatus after it had broken down at 178 deg. cent., but it is highly improbable that he succeeded in obtaining dry and saturated steam above that temperature. He certainly had not obtained it at lower temperatures and apparently was at no time cognizant of that fact. If there was no other source of error present the Regnault circles should be below the correct curve, *i.e.*, the Davis curve should lie above the Regnault circles.

8 At the present state of experimental knowledge of the specific heats of superheated steam the Davis curve seems to be the best that can be drawn. Ultimately, it will probably be found that it should lie somewhat higher at high pressures and it is probable that an equation giving a maximum value of the total heat at a temperature considerably nearer to the critical temperature will represent the facts better than the equation proposed. I have had the opportunity of going over the work of Dr. Davis in considerable detail and in my opinion the claim he makes for his formula of an accuracy within one-tenth of one per cent between 212 deg. and 400 deg. fahr. is justified.

9 For the purpose of facilitating calculations involving saturated and superheated steam two diagrams (Fig. 2 and 3) have been prepared. These diagrams have been plotted using the total heats of saturated steam given in this paper and the calculated corresponding values of entropy and specific volume. The specific volumes, entropies and total heats of superheated steam have been taken or calculated from the best modern data.

10 Of these two diagrams Fig. 2 is the total heat-entropy diagram devised by Professor Mollier, showing the total heat and entropy of steam in any condition and permitting the immediate determination of the work done in the Rankine unjacketed cycle, and of the change in the condition of steam during adiabatic expansion or throttling, and also giving immediate information about wet steam of any usual quality. The other diagram, Fig. 3, is a total heat-pressure diagram showing specific volumes, qualities, and superheats. This diagram is plotted with pressures as abscissæ on a varying scale—equal distances along the axes of abscissæ represent equal increments in the temperature of saturated steam corresponding to the indicated pressures. A scale of this kind has the advantage of spreading out the constant specific volume lines at the lower pressures. The second diagram (Fig. 3) is of the greatest value when problems involving volumes or ratios of expansion are to be solved.

By the use of the two diagrams singly or together it is possible to solve a large number of commonly occurring problems in steam engine and steam turbine work—some of which problems can otherwise be solved only by a protracted series of trials and errors.

11 There is an apparently curious feature about these diagrams to which attention may be called. The lines of constant superheat are seen to diverge from the saturated steam line at high pressures. This of course results from the large specific heat of moderately superheated steam at high pressures. Such a divergence must necessarily take place. The total heat of saturated steam is tending to a maximum at some temperature below the critical temperature; the total heat of superheated steam along a line of constant superheat (and therefore of increasing pressure and temperature) does not pass through any maximum.

PROF. I. N. HOLLIS This, perhaps, is one of the most important subjects that the mechanical engineer has to take up at present. There will not be great changes in Regnault's tables, but the paper points the way to greater scientific accuracy in the work of the mechanical engineer. Up to this time we have had so much work in developing the great projects American engineers have had to undertake, that we have paid more attention to the strictly practical side of questions and have permitted the electrical engineers to surpass us by far in mathematical work given to the profession. So I welcome this paper by Dr. Davis, as it opens the way for greater accuracy in mechanical engineering matters.

PROF. CARL C. THOMAS The discussion during the past few months regarding the properties of superheated steam, and also that which now bids fair to throw additional light upon the total heat of saturated steam, has been exceedingly valuable. The discussion has involved the work of a number of experimenters, and a few points concerning the results of some of the experiments ought to be emphasized lest they be lost sight of or misunderstood. Also, the writer wishes to make a suggestion regarding the proposed revision of the steam tables upon the basis of throttling calorimeter experiments.

2 Knoblauch's experiments begin at about 30 to 50 deg. fahr. superheat for each of the four pressures used; the writer's begin at 18 deg. fahr. superheat for all the pressures used. Knoblauch's upper limit of temperature varies from about 325 to 400 deg. fahr. superheat, while the writer's experiments stop, in all cases, at 270 deg.

fahr. superheat. In that part of the temperature range which is common to both sets of experiments, namely, between 30 or 50 deg. fahr. and 270 deg. fahr. superheat, the results in the two cases are almost identical. The greatest variation appears at about 28 lb. absolute pressure and 54 deg. fahr. superheat where Knoblauch obtains $C_p = 0.478$ and the writer obtains 0.498. The writer's experiments from 18 to 50 deg. fahr. superheat show higher values of C_p than are shown by the extrapolated curves of Knoblauch.

3 Knoblauch worked at absolute pressures of about 28 lb. to 114 lb. while the writer worked at absolute pressures from 7 lb. to 500 lb. Knoblauch obtained his saturation curve by extrapolation from experimental determinations at about 30 or 50 deg. fahr. superheat, while the writer obtained his saturation curve by extrapolation from exper ments made at 18 deg. fahr. superheat. The two curves show widely different values for C_p at saturation, the writer's being much higher than Knoblauch's.

4 In view of the very close agreement in values of C_p from 50 to 270 deg. fahr. superheat, obtained in these two sets of experiments made by entirely different methods, it seems safe to accept either set of these values as substantially correct for this temperature range.

5 Correspondence which has passed between Professors Schröter and Knoblauch and the writer brings out the fact that the former are pushing their experiments into the higher temperature ranges, desiring to corroborate and extend their already published results, which show a minimum value of C_p somewhere in the neighborhood of 230 deg. fahr. superheat for each pressure, followed by an increase of C_p after the region of minimum value has been passed. Their method and apparatus are especially well adapted to experimenting with highly superheated steam. On the other hand, the writer is working back towards the saturation condition, with an apparatus which has been planned with the special end in view of definitely locating the saturation curve. Appreciation of the real difficulties in obtaining exact knowledge as to moisture conditions existing in steam can be attained only by those who have made actual and extensive experiments.

6 The writer's experiments have led him to expect to find a saturation line, and not a "saturated region." Upon this assumption, such curves as are shown in the writer's[1] Fig. 13 should, as shown,

[1] The Specific Heat of Superheated Steam, by Prof. C. C. Thomas, published in vol. 29, Transactions.

pass through the intersection of the coördinates, namely, heat introduced per unit weight of steam, and temperature to which the steam is being superheated. It is with the determination of the form of these curves, near the intersection of the coördinates, that the writer is now engaged. There is no question that a comparatively very large amount of heat is required to cause a very small rise of temperature of dry saturated steam, which means that C_p is comparatively large in value right at saturation. The writer is attempting to ascertain, by the use of very sensitive resistance thermometers instead of the thermo-couples formerly used, whether it is a case of "jogging the steam out of the saturation region," as some writers have suggested, by the introduction of a considerable amount of heat—or whether any small increment of heat will cause a correspondingly very small rise of temperature of steam which has just reached the condition of complete dryness.

7 In the absence of further data at present which would tend to establish the relative correctness of Knoblauch's and the writer's saturation curves, it is interesting to notice that curves can be drawn through Knoblauch's points as given in the writers Fig. 12[1] which will produce the writer's guess at the saturation curve, quite as readily as the one Dr. Knoblauch has made. And, on the other hand, it would be quite possible to use the writer's data as an argument in favor of the accuracy of Knoblauch's curve. However, the writer has some confidence in the general reliability of his curve, because it is based upon experiments extending down to within 18 deg. fahr. of the saturation temperature, while Knoblauch stopped his experiments at a considerably higher temperature. The fact that the two sets of results agree so well, within the temperature-range actually covered in common by the two sets of experiments, affords evidence of the reliability of the values given for C_p; and the further consideration that each of the writer's experiments started with a determination of the saturation point as its basis, and from that went up and fairly met Dr. Knoblauch's results, which were obtained without reference to the saturation point, gives reason for accepting with some confidence the saturation curve marked out by the writer's experiments. However, it will be apparent from the above remarks that the writer is going over this whole question of the saturation line again in the experiments now in progress.

8 The reason why it is worth while to go carefully into this matter is apparent, and is especially cogent at the present time from the standpoint of the engineer as well as the physicist, because of the

interesting suggestion of Dr. Davis that the steam tables based on Regnault's classic experiments be revised on the basis of experiments which have been made on the expansion of steam in throttling calorimeters, or through porous plugs, for the purpose of determining the specific heat of superheated steam.

9 It is not the writer's intention or desire to detract from the interest attaching to Dr. Davis' proposed use of throttling experiments for indicating the extent to which Regnault's values of total heat of saturated steam should be changed. The indications obtained from the throttling experiments will undoubtedly be suggestive and possibly conclusive, but they should not be accepted, without experimental corroboration, as a basis for a revision of the steam tables. During the past eighteen years, to the writer's knowledge, experiments have been made with the object of obtaining values of C_p by throttling steam. The results have never been consistent among themselves, nor have they given values for C_p comparable with those which have been made by the direct means of electrical heating of the steam. For a time some of the throttling experiments were accepted by engineers, and it was then thought that the specific heat of superheated steam was very much higher than it has since been shown to be, the errors in the throttling method all tending to increase the apparent value of C_p.

10 Useful as the throttling calorimeter is, it is not necessary to remind engineers of the very great difficulty of obtaining reliable or at all consistent results as to quality of steam, by its use, no matter how great care is employed as to lagging, position, etc. It may be a question of uniformity of steam conditions at inlet to the instrument, or in the instrument itself at the point where the temperature is taken. The writer is aware that the method by which Dr. Davis proposes to use the results of calorimeter experiments does not involve precisely the same considerations as those involved in determining the quality of steam, or in fact in determining the specific heat of steam by the throttling method. But in view of the well-known troubles that have been experienced in these two lines of experimental work, it seems decidedly inadvisable to base such an important matter as a revision of the steam tables upon experiments in throttling steam, without thorough corroboration by direct measurement.

11 The direct measurement of the total heat of dry steam will be possible of attainment with great accuracy as soon as the saturation line above referred to, and the heat necessary to raise the temperature of unit weight of steam some distance above saturation, have

been definitely and finally determined. Means of applying heat electrically, in absolutely measurable quantity sufficient not only to evaporate water into dry steam but to superheat the steam by some known amount, afford means of direct measurement of the total heat of dry steam, because the heat applied above that necessary to obtain the saturation condition will be known, as soon as the saturation curve has been determined, and this superheat can be subtracted, leaving as a remainder the total heat of dry saturated steam.

12 The writer would not wish it understood that he urges delay in revising the steam tables until his own experiments shall have been completed. That is not at all the point. By the time the saturation curve is definitely located, and possibly before, the writer will be glad to turn the work over to anyone else who may be in position to go on with it; or the desired results may be obtained by some other experimenter entirely independently and before the writer's experiments can be brought to a satisfactory conclusion; but since methods of direct electrical measurement are now available and admirably adapted to this purpose, such means should certainly be used as a check upon the accuracy of the methods proposed by Dr. Davis, before engineers are asked to accept a new set of steam tables. Otherwise we shall have just such a condition of uncertainty regarding the accuracy of our steam tables as has existed for many years with regard to the specific heat of superheated steam.

The Author The friendly words both of approval and of criticism with which this paper has been received are much appreciated by the author. In particular Professor Peabody's intention to recompute his well-known steam tables on the basis of these new values is the pleasantest recognition which they could have. The ease with which this can be done for the wet steam part of his temperature-entropy diagram is most surprising, and it is unfortunate that equally simple laws do not hold for superheated steam. It is hoped that the lack of references which he criticises will be remedied by the accompanying partial bibliography of the subject.

2 The original paper should also have contained a statement as to the heat unit employed. Two such are available, the standard B.t.u., which is the heat required to raise one pound of water from 60 deg. fahr. to 61 deg. fahr., and the mean B.t.u. which is the 180th part of the heat required to raise a pound of water from the freezing point to the boiling-point. Of these, the second is better known in terms of mechanical or electrical units than the first, because the

specific heat of water happens to be changing with temperature more rapidly near 60 deg. fahr. than elsewhere, so that the experimental determination of the standard B.t.u. is difficult and uncertain. An additional advantage of the mean B.t.u. is the simple conversion-factor ($\frac{5}{9}$) between steam tables based on it and those based on the Bunsen or mean calorie, now becoming standard abroad. Inasmuch as the difference between the mean B.t.u. and the 60 deg. B.t.u. is

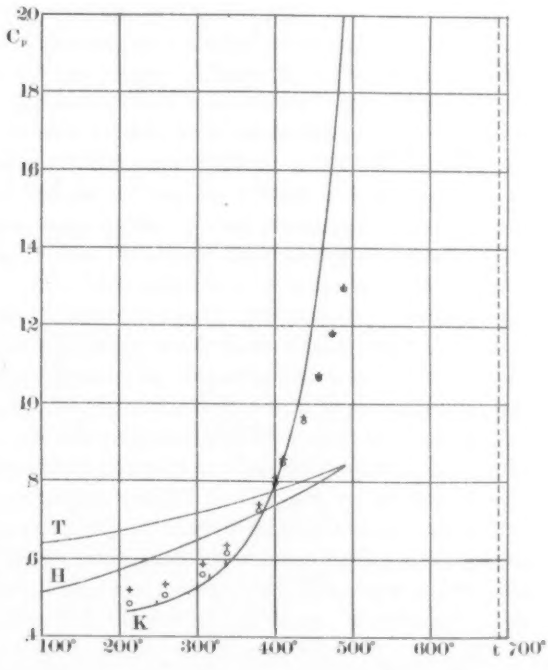

Fig. 1 Reproduction of Fig. 6 of the Author's Paper, with Values Added of C_p from Thomas

probably between one-thirtieth and one-tenth of one per cent, so that it makes practically no difference to the engineer which he uses, the mean B.t.u. has been used in this paper and in the steam tables which are to be based on it.

3 It seems necessary to emphasize again that the point of view of this paper is not so much that the question of the specific heat of steam is "about settled," as that it makes very little difference in

the results of this paper whether it is settled or not, because *the computation method is extremely insensitive to errors in C_p*. This is because C_p comes in as a factor, not in the whole of the total heat (H), but in the small difference between a given H and a standard H_{212}. Now the largest of the 250 computed differences was about 45 B.t.u. If the error in the *mean* C_p from saturation used in computing this particular difference had been 10 per cent too low, the error in the resulting H would have been only $4\frac{1}{2}$ in something like 1200 B.t.u., or just over a third of one per cent. This device of measuring, not the whole of an unknown quantity, but merely the difference between it and a nearly equal standard, is a familiar and a very useful one, as for instance in the accurate measurement of electrical resistances by the Carey-Foster method. As a matter of fact, the actual difference between Knoblauch's mean C_p and Thomas' in the particular case mentioned above was 9.7 per cent, because of the sudden swoop toward saturation of the latter's 15-lb. C_p curve. The use of Thomas' values instead of Knoblauch's would, therefore, make a difference of about 0.36 per cent in H near 400 deg. At 300 deg. the difference would be well under a quarter of one per cent, and it would grow rapidly smaller as one approached 212 deg. As was said in the paper, this is not an estimate of the probable error of the H formula given, for Thomas' values at low pressures and close to saturation are generally admitted to be too large. It is simply to show strikingly how small a difference in H is caused by comparatively large changes in C_p.

4 Finally, the statement that the circularity of the reasoning at the end of the paper is apparent, not real, has been questioned. But the value of H at 212 deg.. and two other values, say at 300 deg. and at 380 deg., "corrected" as above, can easily be used to determine a new second-degree equation for H, and new values of $\frac{dH}{dt}$, of r and of u, *based wholly on Thomas' values of* C_p. A recomputation with these "corrected" data by the method of Par. 19 gives the values of C_p plotted with crosses in the accompanying figure, which is otherwise a reproduction of Fig. 6 of the paper. The circles in it were obtained in the same way from data based wholly on Knoblauch's values of C_p. It is evident that no matter what set of values we start from, the method leads in the end to practically the same curve. The fact that this curve is, in general, much more like Knoblauch's than like Thomas' would seem to be conclusive.

5 The first seven paragraphs of Professor Thomas' discus-

sion are a valuable contribution to the outstanding C_p controversy, especially his statement that according to his new experiments, "there is no question that a comparatively very large amount of heat is required to cause a very small rise of temperature of dry saturated steam." The publication of these new results will be eagerly awaited by all interested in the subject. In the meantime, it should be remembered that this C_p controversy has much less to do with the validity of the results in this paper than might at first be supposed. As has already been pointed out in Par. 3 of my closure, the use of Professor Thomas' values of C_p instead of Professor Knoblauch's would make only a small difference in H. It is interesting to notice that the changed values of H would be even farther from Regnault's than are those proposed in this paper.

6 The last five paragraphs of the discussion raise a much more vital question as to the validity of the throttling experiments on which this paper is based. This criticism has also been made by Professor Heck. It is true that throttling experiments have fallen into disrepute "in view of the well-known troubles that have been experienced" in "two lines of experimental work," namely in determining the quality of wet steam and in computing C_p from H. Of these, the latter is a use for which such experiments are particularly ill-adapted, and it is this very fact which makes the reversal of the process—the computation of H from C_p—so insensitive to errors in C_p. As to the former, one should remember that the ordinary throttling calorimeter, even when "great care is employed as to lagging, position, etc.," is the crudest sort of an instrument of precision, as far as heat insulation and the measurement of the low-side temperature are concerned, so that it is not remarkable that great accuracy is not attained.

7 The experimenters whose results are the basis of this paper used apparatus of a very different sort. Their three different systems of heat-insulation and of thermometry, although by no means perfect, were much better than those of which the average engineer would be reminded by Professor Thomas' allusions to throttling calorimetry. If the precautions of any one of them had not been effective, no such agreement of results based on their work could possibly have been expected as is actually found. The value of their mutual corroboration, in every respect that concerns this paper, is increased by the fact that both Griessmann and Peake were primarily interested in disproving a certain conclusion of Grindley's, so that their critical attitude might have been counted on to ensure substan-

tial improvements in Grindley's results, if that had been possible. If this paper had been founded on Grindley's work alone, the doubts of Professors Thomas and Heck would have had great weight; but the fact that all three pieces of work were used, *and the results agreed*, is good evidence that these throttling experiments are beyond the uncertain and inconsistent stage which Professor Thomas describes.

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[1]Of these only the first had been published when the computations for this paper were being made, but the new experimental data in the second paper agree almost perfectly between 212 deg. and 400 deg. with the values which were used.

No. 1213

FUEL ECONOMY TESTS AT A LARGE OIL BURNING ELECTRIC POWER PLANT HAVING STEAM ENGINE PRIME MOVERS

By C. R. Weymouth, San Francisco, Cal.
Member of the Society

The writer desires to present to the Society the results of various fuel economy tests at the Redondo Plant of the Pacific Light and Power Company, near Los Angeles, Cal., which are of interest both by reason of steam engines having been used as prime movers and because of the notable economy attained. Owing to the rigorous demands on commercially operating power stations, an opportunity is seldom afforded to conduct uniform load tests on a complete plant unit, and the information secured is therefore considered worthy of presentation before the Society.

2 No doubt many members have noted in September issues of some of the leading technical papers a description of the apparatus constituting the Redondo plant. For the better understanding of the results herein presented, a brief statement of the plant make-up will be given.

PLANT DESCRIPTION

3 The station consists of three main units of 5000 kw. each; the layout being in general on the panel system. For each plant unit, there is one McIntosh and Seymour double horizontal and vertical, compound, condensing, automatic, grid-iron valve engine, size 34 and 70 by 56 in., directly connected to an alternator. The rated speed is 100 r.p.m. Each engine has two horizontal high pressure steam cylinders, and two vertical low pressure cylinders and is designed to operate at 175 lb. maintained throttle pressure, with 100

[Data upon California oils and their heat values will be found in the paper by the author upon "Unnecessary Losses in Firing Fuel Oil and an Automatic System for Their Elimination," No. 1214 in this volume—Editor.]

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

deg. fahr. superheat at the throttle. All steam cut-off valves are under control of the one shaft governor. This engine governor is subject to variation in speed, under the control of a McIntosh and Seymour electrically operated speed-changing mechanism, situated within the governor and operated from the switchboard gallery. Each engine is fitted with two superheating receivers designed not only to separate the moisture and oil in the exhaust from the high pressure cylinder, but also to superheat the steam en route to the low pressure cylinders.

4 Directly connected to each engine there is one ATB-60-pole, 5000-kw. 100-r.p.m., 18 000-volt, 50-cycle, 3-phase General Electric generator of the flywheel type. The revolving field is provided with squirrel cage winding to prevent hunting. The generator field rheostats are motor-operated with remote control from the switchboard.

5 In each plant unit are six Babcock and Wilcox boilers of forged steel construction, arranged in three batteries of two each. Five boilers are intended for the nominal capacity of the unit, the sixth boiler for reserve. The boilers are 21 sections wide, 14 tubes high, 18 ft. long, with three 42-in. drums, each containing a total effective water-heating surface of 6042 sq. ft. and designed for 200 lb. working steam pressure. Each boiler is equipped with one Babcock and Wilcox forged steel double-loop superheater.

6 The boilers are provided with Peabody patent fuel oil burning furnaces, having a furnace depth of 10 ft. from boiler front to face of bridge wall. The burner head is placed at the bridge wall, the flame shooting toward the boiler front. The furnace diverges in the direction of the flame, corresponding to the angle of inclination of boiler tubes. Three burners are used per boiler, and are controlled from the boiler front.

7 Corresponding to each main generator, there is one 75-kw. General Electric exciter, driven by means of a 9 and 17 by 12 in. Harrisburg tandem, compound, non-condensing, piston valve, automatic, enclosed, self-oiling engine.

8 There are two Wheeler condensers for each main engine, located in the basement below the operating floor, one condenser being used for each low pressure cylinder. The exhaust steam is led directly from the low pressure cylinder into a 30-in. exhaust main, and after passing through a 90-deg. sweep, drops directly into the condenser. The two condensers on each engine are cross-connected so that both engines can exhaust into either condenser when the other is undergoing cleaning or repairs. Each condenser contains 5002 sq. ft. of

cooling surface, consisting of $\frac{3}{4}$-in. O. D. No. 18 B.W.G. brass tubes. In addition to the regular cooling surface of each condenser, there is provided in the upper compartment a series of tubes comprising a Volz Heater, through which the condensed water is pumped by the air pump at atmospheric pressure. The feed water is thus heated to within a few degrees of the temperature of the exhaust steam surrounding the upper tubes, thus compensating for the cooling action due to the lower condenser tubes.

9 Circulating water is supplied by three large engine-driven centrifugal pumps, connected to a common pump discharge main. Branches from this main supply cooling water to all condensers. The outlets from all condensers connect to a common condenser discharge main. Each condenser has one Edwards triplex, suction, valveless, single action, motor-driven air pump, size 16 by 10 in.

10 After passing through the Volz heaters the condensed water is passed through suitably constructed filters for the elimination of the greater part of the entrained cylinder oil. The outlet of the filters is connected to the feed pump suction main, the supply of water being maintained by an open equalizing hot well, which is also connected to the suction main.

11 Each unit has one Snow duplex, horizontal, boiler-feed pump, having a compound non-condensing steam end and an outside center-packed water end. Each unit also contains one Goubert vertical, closed, auxiliary, feed water heater, of the multiple-flow type, having 1000 sq. ft. of effective tube-heating surface.

12 The condensation in the main engine superheating receiver coils, is led to a simple duplex receiver pump. The hot water, at a temperature of about 360 deg., is returned under boiler pressure to one or more of the boilers in the corresponding unit. The oily drips from receiver bodies are trapped to waste. From the auxiliary measuring tanks, the fuel oil is first pumped through a Goubert closed, multi-flow oil heater, in which the oil is heated to a temperature of approximately 150 deg. fahr., utilizing the exhaust steam from the oil feed pump. From the oil heater the oil is led to an oil pressure main and thence to the oil burners through a suitable system of piping.

13 An automatic system of regulation was employed for the firing of all boilers in the unit tested, hand firing being used on the remainder of the plant. The automatic system controls the supply of oil to the burners, the supply of steam for atomizing purposes, and the supply of air for combustion. This control is obtained through a steam pressure regulator which operates a relief valve in the oil

pump discharge line, and is actuated by the variations in steam pressure in the boiler. This means of control causes a variation in the rate of burning oil comparable with the momentary load. With this system all burner valves may be left wide open or nearly so, and it follows that the intensity of firing increases or decreases simultaneously in all the boilers.

14 The supply of steam to burners for atomizing purposes is from a low pressure steam main, the controlling valves at each of the burners being left wide open or nearly so. The variations in pressure in the low pressure steam main are governed automatically by variations of the oil pressure in the oil main. The regulator used throttles for the supply of live steam to the low pressure main, to produce the desired relationship of oil pressure and steam pressure on the respective burner headers. The supply of air for combustion is automatically regulated by a damper controller, operating a common rock shaft connected to all boiler-outlet dampers. This regulator opens and closes all boiler dampers synchronously with an increase or decrease of oil pressure in the oil main.

15 This system is capable of such operation as to control all of its functions completely and automatically. In actual practice, however, it has been found desirable to watch closely the firing of boilers to make certain that no external conditions such as clogging of burners, distortion of flames by burners, variation in temperature of fuel oil as fired, etc., are present to prevent the attainment of the highest possible efficiency. To minimize heat losses due to radiation, all live steam heat-radiating surfaces are covered with non-conducting covering 3 in. in thickness, and other surfaces are covered in accordance with the dictates of good practice.

OUTLINE OF TESTS

16 The data and results given herein are based on the following tests:

a Official 15-day variable load test on No. 2 plant unit.

b Uniform load test at approximately 2000 kw. output No. 2 plant unit.

c Uniform load test at approximately 3000 kw. on No. 2 unit.

d Uniform load test at approximately 4000 kw. output No. 2 unit.

e Uniform load test at approximately 5000 kw. output No. 2 unit.

f Test complete plant at variable load similar to the official test.

These tests are more fully described under their respective headings.

OFFICIAL FIFTEEN-DAY TEST

17 The official test of this station was on No. 2 plant unit, and was made to determine the economy of the plant under contract conditions, as a basis for computing the bonus earned by the contractor, or the penalty due the owner. The result of this official test is now common knowledge, but not all the conditions surrounding the test have been heretofore published; and as the result of the official test on variable load is of interest for comparison with the results of the uniform load test, herein presented, the writer desires to review the more important conditions attending the official trial. The contract placed the official test under control of a testing committee, the members selected being Prof. C. L. Cory, Dean of the College of Electrical Engineering, University of California, as Chairman; Edward S. Cobb, Chief Engineer for the Pacific Light and Power Company, as Purchaser's Member; and the writer, having charge of the engineering department of Chas. C. Moore and Company, Engineers, as Member for the Contractor (or Company, as herein referred to in extracts from the contract).

18 The Contractor's commercial load economy guarantee was as follows:

The company guarantees * * * that the "First Unit" * * * when operating on commercial railway load, under 90 days test, conforming to the conditions herein specified, will develop an average economy of 170 kw-hr. per barrel of oil.

Duration of test will be 90 days.

The total net output of unit under test for each day will not be less than 60 000 kw-hr., and not more than 78 000 kw-hr.

The main unit under test will be in operation during 19½ consecutive hours per day, and will be shut down 4½ hours per day. The temperature of circulating water entering the condensers will never exceed 70 deg. fahr.

The power factor of the electric load will be at all times between 80 and 100 per cent. Total load on 5000-kw. alternator will never exceed 6000 kw. The load on 5000-kw. generator greater than 5000 kw. will last for an interval not greater than ½ min. duration; and there will be not to exceed 10 such intervals during any hour that the plant is under test.

The load on the unit under test will be such as would be produced by the regu-

lar and normal working of the railway system to which the power is furnished by the purchaser, and varying within the limits herein described.

The load will not be manipulated so as to be either favorable or unfavorable to the showing of unit under test, within the limits herein specified.

19 It was optional with the contractor to insist on the operation of the test unit on a separate transmission line. This would have inconvenienced the purchaser, and for operating reasons a compromise was effected, it being agreed to operate the test unit during the official test in accordance with an agreed load curve, subject to certain upper and lower limits as shown in Fig. 1, in which:

Curve A shows agreed load curve to be followed as closely as possible by electrical operator, as determined by testing committee.

Curve B shows agreed maximum load not to be exceeded by unit during test.

Curve C shows agreed minimum load curve below which the load was not to fall during test.

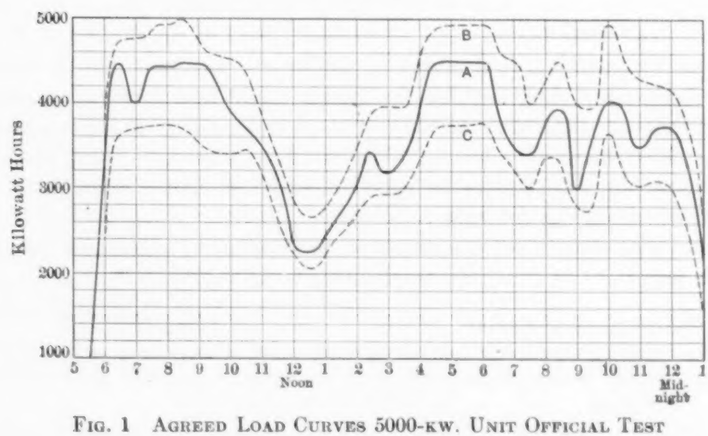


FIG. 1 AGREED LOAD CURVES 5000-KW. UNIT OFFICIAL TEST

CURVE A SHOWS AGREED LOAD CURVE TO BE FOLLOWED AS CLOSELY AS POSSIBLE BY ELECTRICAL OPERATOR AS DETERMINED BY TESTING COMMITTEE; TOTALING 72 280 KW-HR. DAILY OUTPUT. CURVE B SHOWS AGREED MAXIMUM LOAD NOT TO BE EXCEEDED BY UNIT DURING TEST. CURVE C SHOWS AGREED MINIMUM LOAD OR LOWER LIMIT OF OPERATION DURING TEST.

20 The control of the load during test was thus left entirely to the purchaser's station operator within the specified limits and subject to such momentary variations as would be imposed on the test unit by the fluctuation in load on purchaser's system. It was thus

attempted to generate a total of 72 280 kw-hr. per day, with the understanding that should the total kilowatt-hour output up to any hour exceed that shown by the load curve A, then during the remainder of the test period the load was to be modified to correct for any variation in the total output, so as to maintain the total daily output of the unit uniform and as agreed.

21 Endurance features having been established, by reason of the long period of operation previous to the test, the period of test was subsequently reduced from 90 to 15 days. And owing to certain advantages that might accrue to the contractor, such as superior economy due to shortening of test, the contractor agreed to deduct from the bonus otherwise due the sum of $50 000 in consideration of this reduction of test period.

22 Owing to the use of a circulating water discharge main, common to all circulating water pumps, and the steam supply pipes of the circulating pump engines being from a common steam main, it became necessary to eliminate the centrifugal circulating water pump from the unit under test; and in consideration of this omission the contractor rebated the purchaser an agreed amount of the bonus.

23 It was further agreed that the amount of circulating water taken by the two condensers, for the unit under test, should correspond to a rise in temperature of circulating water within the condenser, of between 15 and 18 deg. fahr., such amount of water being controlled by the circulating water outlet valves.

24 In connection with the guarantees, however, it should be stated that the economy guarantee was based on the purchaser's supplying such a siphon system of suction and return circulating water piping, that the total head imposed on circulating pump due to friction and flow of water within pipes, would not exceed a total of 5 ft.

25 The economy guaranteed initially included the provision for lighting the unit under test from energy generated by the test unit. Correction for lighting was made by deducting from the total net output of the unit under test an amount of power equivalent to 125 kw-hr. per day, and the reported economy is based on the net figure after making this deduction.

26 During the test the power for operating the air pump motors, for operating motors connected to circulating pumps used for passing cooling water through engine bearings and guides, for operating electric motor in main engine speed-changing device, and for operating electric motor for controlling switches, was taken from the leads

of the main generator in such a way as to compel the watt meters to indicate only the net useful output of the unit.

27 It is only fair to state that, owing to the layout of the plant, certain auxiliary apparatus in connection with the plant was installed by the purchaser, power for which was not taken from the unit under test; such auxiliaries include the following:

a Electric motor for pumping oil from the main oil storage tank to weighing tanks: the purchaser was compelled to install this electric pump owing to the topography of plant site, the usual gravity feed from main tanks to auxiliary suction tanks being impossible, by reason of the elevation of temporary receiving tanks on weighing scales.

b Electric motor for pumping feed water from the wells, constituting the purchaser's fresh water supply plant, and delivery to the plant under a slight pressure, equivalent to a normal city supply.

c Two small Westinghouse air compressors common to the entire plant, for compressing air, used for cleaning generators, operating certain switches, and forcing cylinder oil from the filtering tanks through a piping system, to save labor of handling by oilers.

d Sump pump common to the entire plant, necessary for pumping overboard waste water from the various sumps; these pumps were installed by the purchaser by reason of the level of sump being lower than sewer.

e Electric motor of 50 h.p. for operating 12 by 12 in. duplex vacuum pump, installed by purchaser to remove accumulation of air in long suction and return circulating water pipes; this motor operated intermittently.

It will be recognized that the auxiliaries omitted, with the exception of the circulating pumps, are not necessarily essential in the generation of power, that is, they may or may not be required in a power plant, depending upon local conditions, etc.

28 During the tests special weighing and receiving tanks were provided for the measurement of the fuel oil required by the unit under test. After weighing, the oil was emptied into a receiving and heating tank, the heating being done by steam from the unit under test, and from there conveyed by gravity to the oil feed pump used for feeding oil to the burners.

29 All instruments were accurately calibrated and standardized. For purposes of the tests a total of twelve integrating watt meters were purchased, and after calibration in place under conditions of operation, it was finally decided to determine the power output by using one watt meter in each phase of the generator, there being three reserve watt meters. Meters were connected in the main winding of the generator between the grounded center of the winding and the winding itself.

30 For the calibration of electrical instruments the testing committee employed Mr. E. F. Scattergood of Los Angeles, an Electrical Engineer and a specialist in such work. There were used certain standards furnished by him, certain standards supplied by the electrical engineering department of the University of California, and certain other standards secured from the Bureau of Standards, Washington, D. C. The oil analyses and heat determinations were made by Prof. Edmond O'Neill, of the department of chemistry, University of California. Check analyses and determinations were made by other chemists.

31 A large corps of assistants and observers were employed during the tests. At one time a total of 80 persons was directly involved. The neutral observers, including the members of the graduating classes in electrical and mechanical engineering of the University of California, were under the direction of the chairman of the testing committee. In all details of the test care and accuracy, in keeping with the importance of the various determinations and the amount of bonus money involved, were observed.

UNIFORM LOAD TESTS

32 During the uniform load tests the same general conditions prevailed as during the official trial, except as herein indicated. Though unofficial, the determination of fuel burned and power output during the uniform load tests was still under control of the testing committee and the same corps of observers, although the watt meter readings were taken less frequently, as a result of experience during the official trial. The calculated economies given are obtained from the readings made by the neutral observers. It should be stated however that the testing committee never completed any official report for these uniform load tests, as the results of the uniform load tests were of no consideration in determining the amount of bonus to be paid the contractor.

VARIABLE LOAD TEST OF THE COMPLETE PLANT

33 During this test the entire plant was operated under commercial conditions of load, except that an attempt was made to follow on all three units the load curve used during the official test. The circulating pumps were supplied with steam from the various boilers, and all plant auxiliaries, including those omitted from official test, were included in the complete plant test. All these last named auxiliaries were not in use at all times, however, but only for the following periods: 10-h.p. motor for water supply pump, in operation during, entire test; 50-h.p. motor for vacuum pump, 16½ hr.; and 10-h.p. motor-operating pump for pumping oil from main to auxiliary tanks, in use during greater part of test. The output of all units was determined by means of station watt meters which have since been carefully calibrated by the purchaser, and the correction factors as found applied to the observed readings.

34 This test did not in any sense do justice to the plant, owing to operating conditions as explained below, and for this reason the test was protested by the writer and at the time was discarded by the testing committee. The writer has since felt, however, that the result is of interest, and offers it with this explanation.

OPERATING CONDITIONS DURING TESTS

35 During all the tests the plant was handled by the regular station operators under the control of the contractor's superintendent of construction, Mr. J. R. Atchison.

36 The load was quite variable, necessitating constant adjustment of auxiliaries, oil burners, dampers, etc. On boilers of No. 2 unit, the automatic system of firing was used during the official test and during uniform load tests. Control of oil burners and dampers on No. 1 and No. 3 units during the complete plant test was by hand.

37 The contractor's experts were present during all tests and cautioned the operators when necessary. Dr. D. S. Jacobus, a member of the Society and Advisory Engineer for the Babcock and Wilcox Company, was present during all the tests, and as manufacturer's representative; Mr. Harte Cooke, a member of the Society, representing McIntosh, Seymour and Company, was also present during all the tests, and much credit is due to both for the exceptional performance of representative apparatus.

38 During the official 15-day test certain unfavorable conditions predominated, for which no correction has been made in the stated

economy. Owing to the accumulation of sea weed in the circulating water pipe line and the interruption of flow of circulating water through the condensers, there were intervals when either one or both condensers of the test unit operated at a reduced vacuum, and at other times it was necessary to run the engine non-condensing. Frequent shorts occurred on the lines, and again periods of excessive overload. During the official test, instead of following the comparative smooth load curve agreed, as shown in Fig. 1, momentary changes in load were of considerable magnitude. Fig. 2, curve A, shows the actual load curve reproduced from the record of the Westinghouse polyphase, curve-tracing, indicating station watt meter, for one day of the 15-day test, being an average day in regard to the momentary variations of load. Actual load curves are similarly given for all uniform load tests. A marked saw-tooth effect will be noted, momentary changes being so violent as to cause unfavorable economy, these variations in load not only affecting the economy of the prime mover, but also the performance of boilers, the continual change in rate of oil firing, air supply, steam supply to burners, etc., making it impossible to secure the best results; although the automatic firing system emplolyed on No. 2 unit no doubt greatly reduced such losses.

39 It will be noted that certain of the uniform load tests were of comparativey short duration. The results of such tests are therefore subject to slight error. The initial period of starting the uniform load tests was determined by the testing committee, and while the writer has reluctantly assented to certain of these intervals, it will be self-evident that for certain of these tests the period allowed for warming up boilers before starting of test was not sufficient to eliminate all of the heat storage effect, and is not fully in accordance with the Society's rules for conducting such trials. Previous to certain tests, the boilers had been down for such lengths of time as to be practically cold. In other cases the boilers had been at stand-by for four or more hours previous to starting fires; except during the 3000 kw. test where the boilers after previous day's run were kept up to steam pressure by intermittent firing. The results, therefore, from this standpoint are slightly unfavorable, the fuel consumption stated involving to some extent the fuel loss due to warming up the boilers.

40 The test for the 2000-kw, load was made with three instead of five boilers in operation, as is the regular practice at this plant during any prolonged running at such load. Owing to the short period of the test, it would have been impossible to keep the two idle boilers on the line and fire at intervals during the test, in such a manner as to

guarantee the same amount of heat storage in the idle boilers at both commencement and end of the test period. The amount of such loss due to stand-by of the idle boilers was therefore determined separately by actual measurement, and the resulting fuel loss is given in the accompanying table, corresponding to which due correction in economy has also been figured. The amountof this correction corresponds to the heat necessary to keep the two idle boilers continuously up to full boiler pressure. It is a fact, however, that the actual stand-by loss was less than this amount, owing to the reduction in the rate of heat radiation which takes place after the boiler has appreciably cooled.

41 During the uniform load tests it was necessary, owing to peculiar conditions of operation, to lower materially the power factor of the unit under test, in order to control the load on same more readily. Owing to this lowering of the power factor below the range for which the watt meters were carefully calibrated, an error was introduced in the readings, manifesting itself in a tendency of the meters to run too slow, thus indicating an output less than actual. Further, owing to the lower power factor imposed on the main generator, the C^2R and other losses were greater than normal (80 to 100 per cent), causing a falling off in generator efficiency and consequent reduction of fuel economy.

42 During the complete plant test the head on the circulating pumps was measured and averaged in the neighborhood of 35 ft. This excess head is due to the fact that the purchaser met with certain reverses in the installation of the circulating water pipe line, making it necessary to install internal-angle iron strengthening ribs. The suction and discharge piping is also subject to certain air leakage, and to a considerable accumulation of sea weed, both in the main suction line and in the suction line strainers. As a result of this condition, not only were the total head and duty imposed on the circulating pumps during the complete plant test exorbitant, but the quantity of circulating water handled was not quite sufficient for the most economical vacuum requirements when all three units were in operation.

43 For periods during the complete plant test, the vacuum was very poor indeed, owing to the accumulated sea weed in the suction line and condensers faster than the strainers could care for it. During all these tests, certain unfavorable conditions existed as to the character of oil burned and the treatment of same preparatory to burning, as explained more fully below under the caption, Fuel Oil Specifications and Corrections.

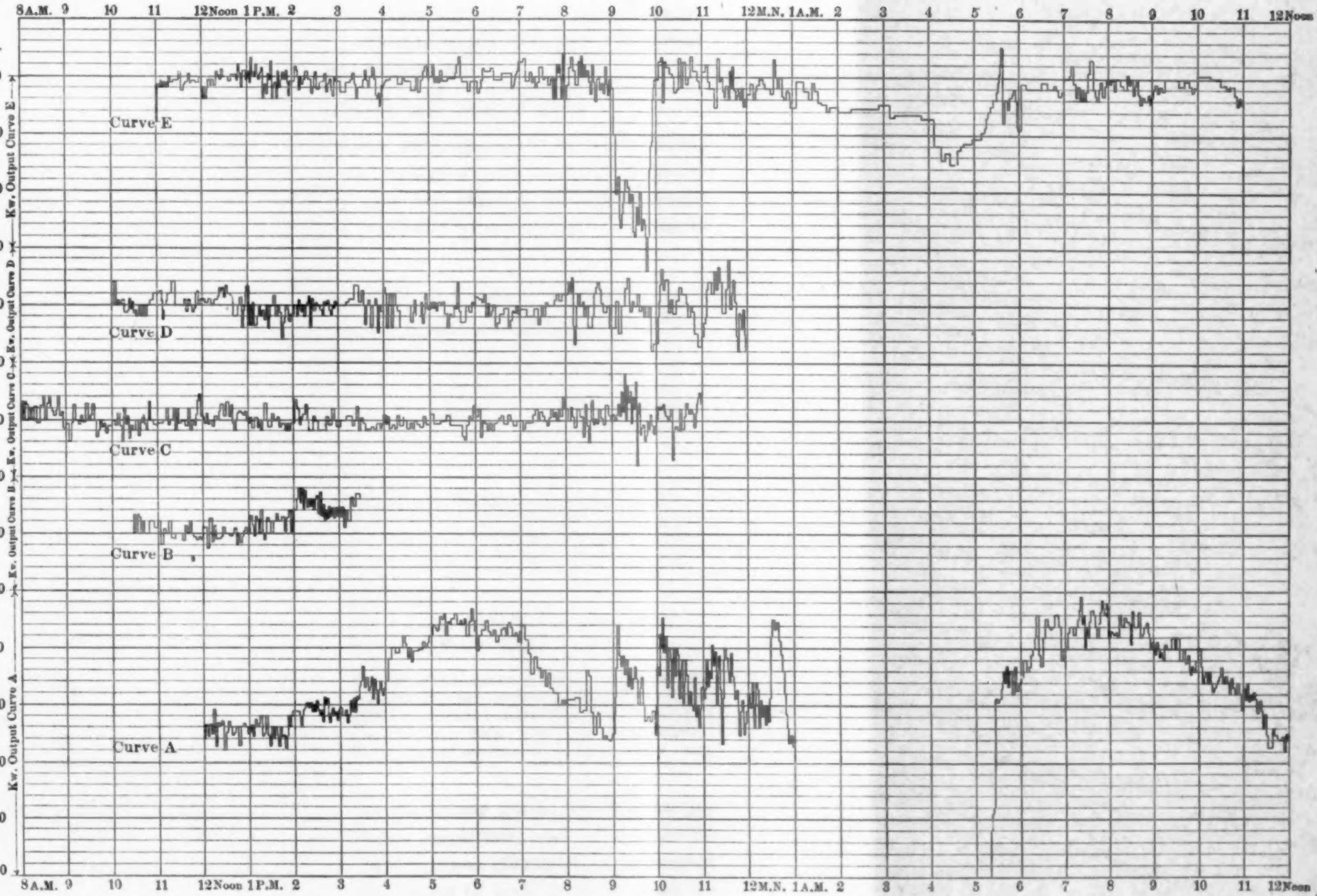


FIG. 2 LOAD CURVES FOR UNIFORM AND VARIABLE LOAD TESTS

CURVE A SHOWS THE LOAD VARIATIONS FOR THE ELEVENTH DAY OF THE 15-DAY TESTS, 12 NOON TO 12 NOON. CURVE B SHOWS THE LOAD VARIATIONS FOR THE TEST AT APPROXIMATELY 2000-K.W., 10.30 A.M. TO 3.30 P.M. CURVE C SHOWS THE LOAD VARIATIONS FOR THE TEST AT APPROXIMATELY 3000-K.W., 8 A.M. TO 11 P.M. CURVE D SHOWS THE LOAD VARIATIONS FOR THE TEST AT APPROXIMATELY 4000-K.W., 10 A.M. TO 12 M. CURVE E SHOWS THE LOAD VARIATIONS FOR THE TEST AT APPROXIMATELY 5000-K.W., 12 NOON TO 11 A.M.

OBSERVED DATA

44 In Table 1 is given a summary of all the observed data and the calculated results for the various tests, including averages of steam pressure at engine throttles, superheat at engine throttles, condenser vacuums, circulating water temperatures, etc.

FUEL OIL SPECIFICATIONS AND CORRECTIONS

45 Certain arbitrary corrections were agreed on in formulating the guarantee of this plant, and the nature of such corrections should be well understood in order to gage the value of the results herein given and the possible error that may have been introduced by reason of such corrections.

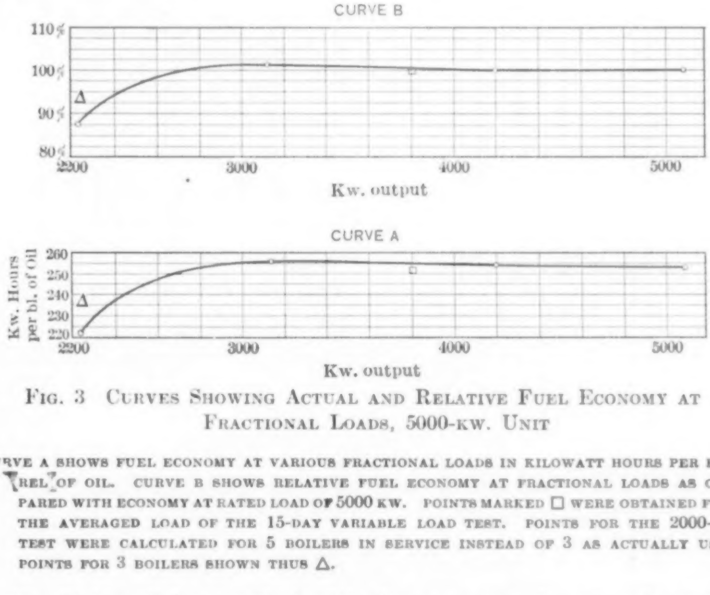


FIG. 3 CURVES SHOWING ACTUAL AND RELATIVE FUEL ECONOMY AT FRACTIONAL LOADS, 5000-KW. UNIT

CURVE A SHOWS FUEL ECONOMY AT VARIOUS FRACTIONAL LOADS IN KILOWATT HOURS PER BARREL OF OIL. CURVE B SHOWS RELATIVE FUEL ECONOMY AT FRACTIONAL LOADS AS COMPARED WITH ECONOMY AT RATED LOAD OF 5000 KW. POINTS MARKED □ WERE OBTAINED FROM THE AVERAGED LOAD OF THE 15-DAY VARIABLE LOAD TEST. POINTS FOR THE 2000-KW. TEST WERE CALCULATED FOR 5 BOILERS IN SERVICE INSTEAD OF 3 AS ACTUALLY USED. POINTS FOR 3 BOILERS SHOWN THUS △.

46 It is well known that with the poorer grades of coal, having excessive quantities of ash, sulphur, moisture, and other volatile matter, there is a falling off in the furnace efficiency as compared with the combustion of the better grades of coal; so with crude oil having excessive quantities of moisture, sulphur and silt, or of low specific gravity, there is an appreciable falling off in the furnace efficiency as compared with a grade of crude oil of average quality. It is certain however that most of the losses due to foreign matter are less with oil fuel than with coal, but the exact extent of these losses is so far unknown.

TABLE I SHOWING RESULTS OF VARIABLE AND UNIFORM LOAD TESTS ON ONE 5000-KW. UNIT, ALSO VARIABLE LOAD TEST ON COMPLETE PLANT OF PACIFIC LIGHT AND POWER COMPANY, REDONDO, CAL.

	Designation of test	Unit	Variable load average of 15-day tests	2000-kw. load test (Approximate)	3000-kw. load test (Approximate)	4000-kw. load test (Approximate)	5000-kw. load test (Approximate)	Variable load complete plant test
2	Date of starting test	1908	April 18	May 14, 10:30	May 19, 8 a.m.	May 18, 10 a.m.	May 5 12 noon	May 21, 11 a.m.
3	Date of stopping test		May 4	May 14, 3:30	May 19, 11 p.m.	May 18, 12 m. n.	May 6, 11 a.m.	May 22, 11 a.m.
4	Duration of test	Hours	24 hr. each day	5	13	14	23	24
5	Time of starting fires	....	....	5:10 a	3:00 a.m.	5:10 a.m.	4:35 a.m.	5:05 a.m.
6	Period of warming boilers	Hours	....	5:20	5:0	4:50	7:25	:55
7	Average steam pressure at engine throttles	Lb. per sq.in.	180.03	183.	181.74	180.9	189.3	173.76
8	Average superheat at engine throttles	Deg. F.	82.46	87.99	92.	96.05	92.78	95.15
9	Average temperature circulating water inlet	Deg. F.	63.03	61.64	61.41	61.34	62.4	59.33
10	Average temperature circulating water outlet	Deg. F.	79.15	78.09	79.37	79.02	81.09	82.12
11	Average vacuum in condenser (Corresponding 30 in. Bar)	In. Hg.	28.334	28.426	28.343	28.214	27.976	27.784
12	Average temperature of feed water leaving heater	Deg. F.	146.9	184.3	167.8	155.9	150.7	177.22
13	Kilowatt output (including lights)	Kw-hr.	71615.24	11225.577	47126.457	58745.125	116899.748	215262.438
14	Deduction for lights	Kw-hr.	125.	26.042	78.125	79.917	117.792	....
15	Net kilowatt output, deducting lights	Kw-hr.	71490.24	11199.535	47048.332	58665.208	116781.956	....

TABLE I.—*Continued.*

16	Fuel oil as fired (334 lb. to bbl.)	Bbl.	303.387	50.01	195.458	244.783	496.910	957.566
17	Heat units per pound oil as fired	B.t.u.	17840.	17938.8	17920.8	17965.8	17838.	17717.
18	Sulphur in oil (by weight)	Per cent	2.34	2.17	2.43	2.39	2.49	2.60
19	Moisture in oil (by weight)	Per cent	2.38	1.82	2.08	1.895	2.70	2.59
20	Silt in oil (by weight)	Per cent	.14	.138	.14	.113	.10	....
21	Fuel oil corrected as per contract	Bbl.	282.746	47.219	183.307	230.764	460.884	883.115
22	Economy (oil corrected as per contract)	Kw-hr. / Bbl.	252.842	237.298	256.664	254.252	253.382	243.758
23	Economy (oil corrected as per contract)	B.t.u. / Kw-hr.	24438.	26039.	24074.	24302.	24386.	25349.
24	Economy corrected only for heat units in oil	B.t.u. / Kw-hr.	25288.	26742.	24857.	25027.	25347.	26320.
25	Number of boilers in service	No.	5	3	5	5	6	15
26	Additional fuel oil 2000-kilowatt test account idle boilers equivalent 5 boilers in use	Bbl.	....	50.339	....	....	....	....
27	Calculated economy 2000-kilowatt test equivalent 5 boilers in service (oil cor.)	Kw-hr. / Bbl.	....	222.482	....	....	....	....
28	Combined efficiency of engine and generator based on separate excitation	Per cent	....	90.2	92.5	94.1	94.75	....

47 The writer would explain that during the preliminary negotiations for the Redondo plant the contractor's guarantee of 170 kw-hr. per barrel of oil was based on an average quality of Bakersfield crude oil, of known furnace performance, having the following specifications:

Specific gravity, 16 deg. Baume.
Corresponding weight per bbl., 336 lb.
Heat units per lb. of oil calorimeter test, 18 500.
Moisture limit, 1 per cent.
Sulphur limit ½ per cent.

48 On the day of closing the contract the purchaser insisted on testing the plant with oil secured from the purchaser's nearby wells; and while this oil never had been used during any authentic boiler trials, yet it was known to be of low gravity, and low in heat units, to have considerable quantities of both moisture and sulphur, and in general to be a decidedly inferior grade of oil.

49 After some discussion a compromise was affected, whereby the economy guarantee was allowed to remain as stated, the oil specifications being changed to the following:

Minimum specific gravity of oil, 14 deg. Baume.
Maximum specific gravity of oil, 18 deg. Baume.
Agreed weight per bbl., 334 lb.
Minimum heat units per lb. of oil, 17 000.
Moisture limit, 5 per cent.
Sulphur limit, 2½ per cent.
Silt, etc., limit, ½ per cent.

50 It was finally agreed that in addition to correcting on the heat unit basis a further correction should be made for the presence of sulphur exceeding the limit of 1 per cent or moisture exceeding ½ per cent, and the purchaser should be penalized by deducting as oil from the weight of oil fired the weight of all moisture in excess of ½ per cent, all sulphur in excess of 1 per cent and all silt and other foreign matter.

51 It is a well known fact that oil, low in gravity, requires an excess of steam for preheating in the auxiliary suction tanks, both for the preliminary treatment of the oil in an attempt to remove the excess of moisture and silt by warming and also to enable the oil pressure pumps to lift this oil, under suction without danger of interruption in the pumping of oil to the burners. It is also well known

that all crude oils are complex mixtures of various hydro-carbons, that distillation takes place to varying extents at all temperatures, and that the higher the temperature to which the oil is heated in an open tank, the greater the loss due to such distillation. The writer had tests made by Mr. C. H. Shepard, chemist, of San Francisco, to determine as near as may be the loss due to distillation caused by such preheating in the case of the Redondo tests and his report is as follows:

Regarding the loss by distillation in open tank at temperatures of 120 deg. fahr. and 130 deg. fahr., you stated in your letter that you were using about 100 000 lb. of oil per day and that the diameter of your tank is about 10 ft. I assume that you would have about 40 000 lb. of oil in the tanks. If the oil were flowing in and out at the same rate any particle of oil would take at least eight hours to pass through the tank. I have kept one sample for eight hours at 120 deg. fahr. and another for eight hours at 130 deg. fahr. in open vessels 4 in. in diameter and oil to a depth of about 3¼ in. without agitation. Results are as follows:

TABLE 2

	At 120 deg. fahr. for 8 hr.	At 130 deg. fahr. for 8 hr.
Total loss in weight	1.53%	1.95%
Loss of water	.35%	.59%
Loss of oil	1.18%	1.36%
Heat unit loss at 20 000 per lb	.236 B.t.u.	.272 B.t.u.

Per cent in water in original sample was by distillation 2.75 per cent.

52 It is further well known that the heavier the oil, the greater the amount of heating necessary in the pressure heaters between pumps and burners, where heat is added to facilitate atomization of the oil and to economize in the amount of steam necessary for such atomization. In addition to the general falling off in furnace efficiency due to the poorer grade of crude oil, there was a further and rather indeterminate loss due to the effect of sulphur and silt, which caused a slight clogging and corrosion of the burners and at times a distortion of the flames. The sulphur fumes from the products of combustion also escaped from the boiler setting, causing a serious annoyance to the firemen and necessitating an excess of air supply to overcome the trouble. With the particular setting of front and rear dampers for greatest economy and minimum air supply, there was at the top of the first pass inside of the boiler setting, a positive pressure in lieu of a draft, due to the lighter column of very high temperature

gases within the boiler setting, as compared with the weight of the outside air at the boiler room temperature. Sulphur was present in the oil to such an extent that the firemen were seriously annoyed unless an excess of air was admitted, increasing the draft within the boiler setting and preventing the escape of sulphurous fumes, thereby causing the well recognized loss in economy due to excess air.

53 With our present knowledge of the combustion of oil fuel, it is impossible to rationalize the results of the Redondo tests so as to determine exactly the true thermal efficiency that would have been attained had an average grade of crude oil been used. During the tests, as will be noted from the values given in Table 1, the grade of oil used was of very low specific gravity and contained excessive quantities of moisture, sulphur and silt. It is indisputable that an appreciable loss in efficiency resulted from the presence of these foreign substances, but owing to the arbitrary nature of the corrections made the plant economies herein stated may be in error by small percentages, the amount of which must remain indeterminate. This criticism will likewise apply to all similar tests which are not made with a grade of fuel recognized as standard.

54 As a matter of interest, the writer has included in Table 1 figures showing the plant efficiency omitting the contract corrections; it must be borne in mind, however, as above stated, that these figures evidently represent a lower economy than would have been obtained with the standard grade of oil specified.

55 Considerably better economy would be expected in the uniform load tests than was secured in the official test, namely 252.842 kw-hr. per barrel of oil. The actual results were not very much higher, however, for as already explained under the heading of Operating Conditions During Tests, the period allowed for warming up the boilers before starting certain of the uniform load tests was not sufficient to overcome the loss due to radiation during the lay-over period and the results were lower than they should be on this account. It should also be borne in mind that in the so-called uniform load tests the actual load varied from time to time, requiring a constant adjustment of oil burners, dampers and auxiliary apparatus, and while the magnitude of these variations was small yet the variation in load was such as to cause a loss in economy as compared with an absolutely uniform load.

56 The economy obtained at the various uniform loads is plotted in the curve shown in Fig. 3. The economy at various fractional loads as compared with the economy at the rated load is also shown in Fig. 3.

57 Inasmuch as the absolute economy at the various uniform loads is probably greater than reported, for reasons pointed out above, and as the correct economy at the uniform load tests would probably be a small but uniform percentage greater throughout all of such tests, it is fair to assume that the relative economies as shown are reasonably accurate, and for this reason the writer considers the curve showing the relationship of economies at various loads to be of greater value than the results of individual tests. There has also been plotted the points corresponding to three in lieu of five boilers, for 2000-kw. load, for both of the above curves, as well as the economy obtained for the official test for the equivalent average load.

58 In studying these curves it should be borne in mind that the results given indicate the entire falling off in fractional load efficiency for the complete plant, being the multiplied effect due to various losses, such as electrical efficiency of generator, mechanical efficiency of engine, indicated economy of engine, auxiliary performance, and the greater effect at light load of dead load losses, such as heat radiation from exposed piping and other radiating surfaces, and the loss in efficiency of boilers owing to operation at fractional loads. As the total effect of all these losses is relatively small, it is in itself evidence of the high maintained steam economy of the prime mover at very light loads. The combined efficiencies of engine and generator are shown in Table 1 and as this falling off in efficiency at light loads is quite apparent, it offers additional evidence as to the maintained indicated economy of engine at fractional loads.

59 A falling off in economy would be expected for the complete plant test of all three units as compared with the official test of No. 2 unit for many reasons; among these are the excess of head imposed on circulating pump on account of pipe line, for reasons explained above; the deficiency in quantity of circulating water, causing a severe loss of vacuum at intervals, and at the completion of test the operation of the plant non-condensing; the use of energy from the plant to operate a 50-h.p. motor driving a 12 by 12 in. duplex vacuum pump, necessary to remove air leakage in suction line; the use of energy in pumping water from the plant for operating motor-driven oil pumps in pumping oil from main oil tanks to auxiliary oil tanks; the use of energy from plant in pumping well water from the purchaser's wells to the plant; the energy used for pumping out sumps; the energy used for lighting various surrounding buildings and wharf; the necessity for hand control of firing the boilers of No. 1 and No. 3 units, the automatic system being in use on No. 2 unit only. Con-

sidering all of these conditions, the economy for the complete plant is very favorable indeed.

60 As having a bearing on the economy of the plant, the writer would explain that to conform to the contract load requirements during test, which are designated as the maximum requirements of the plant during commercial operation, and further to economize in the first cost of plant, the main engines were initially selected to have a normal rated capacity of 4000 kw. in lieu of 5000 kw., the rated capacity of generators. The writer hoped thereby to obtain sufficiently increased economy at the lighter loads to offset the possible loss in economy due to overload. It has been shown that these 4000-kw. engines are capable of carrying overloads in excess of 7000 kw. It appears, however, that it would have been a decided improvement had larger engines been selected, conforming more nearly to the rating of the generators. Due to the use of superheated steam, high fractional load economy would have been insured, the overload capacity still further increased, and superior economy maintained at loads considerably above rating.

61 It is regrettable that the circulating pumps were omitted from the official and uniform load tests. Had the circulating water system conformed to the contract stipulations, or had circulating water conditions existed as in many eastern central stations, where parallel suction and discharge conduits are run directly underneath the circulating water pump, the total head on the system would have been very small indeed. As stated above, the contract stipulation for loss of head in the piping was 5 ft. Had it been possible to conduct the official 15-day test with the circulating pumps, operating under this head, it is safe to say that the additional fuel consumption to operate the circulating pumps would have been scarcely noticeable. The circulating pump engines are of compound non-condensing type and designed for use with superheated steam, insuring a low steam consumption. Inasmuch as the boiler feed water temperature was at all times considerably under 200 deg. fahr., and as practically no exhaust steam was wasted at any period of operation, it is safe to state that all the heat in the exhaust steam from the circulating pump engines would have been returned to the system; compensation for greater steam consumption manifesting itself in a higher feed water temperature.

62 Having no bearing on the economy of the unit, but as a matter of particular note, the writer would state that following the acceptance test on the plant, the engine for the No. 1 unit was operated under

various conditions to determine the possibilities in case of an emergency. It was shown that the unit could be operated on either one, two or three as well as four cylinders. The running conditions were smooth, and the successful parallel operation of the alternators was easily accomplished under all conditions. It was further possible temporarily to overload the cylinders in use during these special trials, to such an extent as to produce in the neighborhood of 1800 kw. per cylinder. The time required to remove a connecting rod in case of an emergency was found to be between one and two hours. It was also found possible to operate the entire engine on one condenser, as well as both condensers, causing but a slight reduction in vacuum, the second condenser being available for repairs, etc. In case of difficulty through sea weed in the circulating water the engines were often run for a long time non-condensing.

63 The writer believes that this type and arrangement of prime mover offers features of flexibility that merit very careful consideration, and enables the maintained operation of steam power plants, under both normal and emergency conditions, with a far less percentage of reserve capacity than is permissible with any other type of prime mover. The Redondo Plant is today operating without any reserve capacity whatsoever, and since the completion of the work it has regularly operated under all conditions of service desired by the management.

64 As the maintained economy of any type of prime mover is of great importance, the writer is able to state that the station records so far fail to indicate any considerable falling off in economy as compared with test conditions. On the other hand, it is a matter of record that a complete plant economy has been attained, since operation of plant by purchaser, exceeding that herein reported. During the dry season this year, and by reason of the large overload capacity afforded by these engines, the management has been compelled, and has been able to operate this station under continuous load far in excess of that for which it was designed. The engines have been operated for long periods at over-loads of 75 per cent above the rated loads, and during such periods only a slight falling off in economy has resulted, which was less than would be expected on account of the special way in which the engines were rated.

65 No. 1 and No. 3 units are now being equipped with an automatic system of firing similar to that on No. 2 and it is expected that the station economy will be materially improved thereby.

COMPARISON OF ECONOMY WITH COAL BURNING PLANTS

66 For the purpose of comparing the economies herein reported with those obtained in modern coal burning stations, the writer has given in Table I the fuel consumption reduced to the corresponding B.t.u. per kw-hr.; these figures are possibly subject to slight error however by reason of oil corrections already explained.

COMPARISON WITH TURBINE PERFORMANCE

67 To the writer's knowledge, the best turbine performance on the Pacific Coast based on oil fuel is considerably under that found for the Redondo Plant.

68 When comparing these results with economies reported for Eastern turbine stations, the writer would call attention to the necessity of making due allowance for the variation in the initial temperature of the circulating water. The average temperature obtainable in the vicinity of Los Angeles is from 60 to 70 deg. fahr., as compared with circulating water temperatures in the neighborhood of the freezing point obtainable at times in many Eastern stations, this lower temperature enabling the attainment of a considerably improved vacuum. Inasmuch as the steam turbine gains largely in apparent economy due to increase in vacuum, similar reductions should be made in the stated economy of such plants, and correspondingly in the reported fuel consumption.

69 The writer would take this opportunity to urge on the Society the establishment of a standard of performance for comparison of fuel economies of complete plants, to be based on the Rankine cycle, the upper limit being the maximum boiler temperature and pressure, the lower limit for all plants operating condensing, being the initial temperature of circulating water, in lieu of the temperature of engine exhaust. This would be an extension of the standard of performance established by the British Institute of Civil Engineers for the comparison of steam engine performance, in which the Rankine cycle is accepted as ideal, the upper limit being based on the throttle steam temperature and pressure, the lower limit on the temperature corresponding to the engine exhaust.

CONCLUSION

70 The writer believes that the performance herein indicated is noteworthy, and warrants on the part of designing engineers a far more careful and thorough investigation as to the capabilities of steam engines, when used as prime movers, than is manifest in the practice of late years.

Discussion on this paper will be found at end of paper No. 1214.—The Editor.

No. 1214

UNNECESSARY LOSSES IN FIRING FUEL OIL AND AN AUTOMATIC SYSTEM FOR THEIR ELIMINATION

By C. R. Weymouth, San Francisco, Cal.

Member of the Society

Practically all oil-fired boiler plants in stationary practice are subject to hand control throughout. It is customary to maintain a uniform oil pressure at the oil pump and in the oil pressure main, and to throttle the supply of oil by hand at all of the individual burners. It is also customary to operate with full boiler steam pressure on the main suppplying steam to all the burners, and to regulate by hand the supply of steam for atomizing purposes, at each of the individual burners. Boiler dampers also are all subject to hand control on the individual boilers.

2 In a central station having say twenty 500-h.p. boilers there would be about 60 burners. For economy of labor, there would probably be not more than two or three firemen to the shift, in a plant of this size. On a commercial railway or lighting load subject to the usual fluctuations, such a plant would probably be operated with the rear boiler dampers clamped in fixed positions, wide open or nearly so. The supply of steam to the burners would receive little attention, but the supply of oil to the burners would be regulated for variations in load by throttling to the extent necessary for maintaining the desired steam pressure. In such a plant there would be a more or less uneven rate of firing at the various boilers, and an excess of air for combustion at all loads, particularly at the lighter ones corresponding to a nearly uniform rate of flow of air through the furnace. The operators are likely to become careless, not noticing the drop in steam pressure with a sudden increase in load until this has become considerable, necessitating a severe momentary rate of firing in a number of boilers to bring the steam pressure back to

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

the normal. This severe duty increases the expense for repairs to the boiler settings, rate of burning out of tubes, etc.

3 In certain plants where engineers are enlightened as to the principles of combustion, the attempt is frequently made to operate on a reduced air supply, with the result, if the dampers are set for mean or nominal load, that the chimneys smoke excessively on overloads before the limited number of firemen can reach all the dampers to open them.

4 As the lamentable result of these conditions, the average boiler plant efficiency with crude oil, even with the best types of boilers, averages much nearer 70 than 80 per cent, which is possible in large plants under proper methods of control.

5 Probably it will always be impossible to instil into the mind of an ordinary fireman such knowledge of the principles of combustion and the losses due to excess air supply, as to obtain economical results in large stations where it is necessary to depend on hand firing. Improved conditions can be secured by the employment of a boiler room engineer whose duty it is to scrutinize all fires from time to time and to coach the firemen in their duties; but the only ideal method seems to be an automatic system of control, such as will be here described, where the various adjustments, having once been made for economical conditions, are automatically repeated for the various conditions of load, maintaining a high average economy from month to month. With well-designed oil furnaces and careful adjustment under uniform load conditions, carefully conducted tests have shown that it is possible to obtain high percentages of CO_2, indicating as low as 10 per cent excess air over the requirements for perfect combustion, with no unconsumed elements in the flue gases.

6 Numerous data relating to coal fuel are available, showing the importance of reduced air supply as tending to high furnace efficiency; also the relation of excess air supply to any observed percentage of CO_2 and other factors of gas analysis. As few data for oil fuel are available, the following will be presented.

7 All Pacific Coast crude oils contain a certain amount of moisture, sulphur, nitrogen and oxygen; the main constituents being carbon and hydrogen. The characteristic difference in oils of different gravities lies in the relative quantities of carbon and hydrogen contained, there being more carbon and less hydrogen in the heavier oils, less carbon and more hydrogen in the lighter. In the better grades of oils treated at the wells before shipment, in which moisture has been largely eliminated, it can be roughly assumed that 3 per

cent of the oil is made up of sulphur, nitrogen, oxygen and water. This relationship is not universal, certain Southern California oils containing a large percentage of sulphur.

8 The predominating oil used on the Pacific Coast, known as Bakersfield oil, averages about 16 deg. Baume; which is equivalent to 336 lb. of oil per 42-gal. bbl. The ultimate analysis of this oil is about as follows:

Carbon, 85 per cent.	Nitrogen, 0.2 per cent.
Hydrogen, 12 per cent.	Oxygen, 1 per cent.
Sulphur, 0.8 per cent.	Water, 1 per cent.

A number of lighter oils in general use, ranging in the neighborhood of 18 to 20 deg. Baume, would average about as follows:

Carbon, 84 per cent.	Nitrogen, 0.2 per cent.
Hydrogen, 13 per cent.	Oxygen, 1 per cent.
Sulphur, 0.8 per cent.	Water, 1 per cent.

Certain heavier oils ranging from 12 to 14 deg. Baume average about as follows·

Carbon, 86 per cent.	Nitrogen, 0.2 per cent.
Hydrogen, 11 per cent.	Oxygen, 1 per cent.
Sulphur, 0.8 per cent.	Water, 1 per cent.

9 As a result of tests by Edmond O'Neill, Professor of Chemistry of the University of California, the calorific value of Bakersfield oil may be taken as about 18 600 B.t.u per lb., allowing for the presence of about 1 per cent moisture as indicated above. When corrected for moisture the net heat units per pound of oil are proportionally higher, although there is a slight loss in furnace efficiency due to the presence of moisture, inasmuch as all such water is evaporated into steam and superheated to the temperature of the escaping gases, involving an amount of heat both sensible and latent.

10 On the basis of the above analyses, the chemical requirements of air for complete combustion per pound of oil are as shown in Table 1.

11 In the various text books, the values given range from 16 to 18 lb. of air per lb. of oil, but an average of 14 lb. of air per lb. of oil is more nearly correct.

12 The ordinary method of indicating and measuring steam to atomize oil has been to express the quantity as a percentage of the

actual amount of water evaporated in the boiler. This percentage ranges from about 2 to 5 and over, depending on the system of oil burning, type of burner, etc. While such a percentage rating is no doubt convenient, it is inaccurate, in that the steam consumption of oil burners is proportional to the oil burned and not to the water evaporated. Various tests have shown that the steam consumption of oil burners ranges from 0.14 to over 0.5 lb. of steam per lb. of oil. The average value of good performance is about 0.3 lb. of steam per lb. of oil, although with hand regulation on variable load this quantity should be slightly increased, and is somewhat dependent on the gravity of the oil, temperature at the burners, etc. In stationary practice, the use of air for atomizing purposes has been practically aban doned.

TABLE 1 WEIGHT OF AIR REQUIRED FOR COMBUSTION OF OIL OF DIFFERENT GRADES

Grade of oil	Light	Medium	Heavy
Per cent of C	84.00	85.00	86.00
Per cent of H	13.00	12.00	11.00
Per cent of S	0.80	0.80	0.80
Per cent of N	0.20	0.20	0.20
Per cent of O	1.00	1.00	1.00
Per cent of H_2O	1.00	1.00	1.00
Air chemically required per pound of oil, calculated—pounds	14.25	14.02	13.79
Corresponding maximum CO_2 by volume, in dry gases of combustion—per cent	15.16	15.52	15.89

13 As no direct experiments have been made showing the loss in boiler efficiency due to various percentages of excess air supply, the writer will present some simple calculations showing the amount of this loss.

14 It is well known that the loss due to an excess of air supply is not only on account of the direct loss in heating the added air to the temperature of the flue gases, but there is a secondary loss due to the fact that corresponding to an excess of air, there results a higher flue temperature not only for the actual amount of air necessary for combustion, but for all such excess air. Calculations as to boiler performance are simplified with oil fuel, as practically complete combustion is secured, in a well designed furnace, the carbon and carbon monoxid usually being burned to CO_2. The stack losses include the sensible heat contained in the dry gases of combustion, the sensible and latent heat in the steam from the combustion of hydrogen and

oxygen and in the steam introduced through the burner, and the moisture present in air for combustion.

15 Assuming complete combustion, and employing a boiler radiation loss of 3 per cent, the writer has calculated the boiler efficiency, at rating, for various percentages of excess air supply, as given in Table 2.

TABLE 2 BOILER EFFICIENCY FOR EXCESS AIR SUPPLY

Excess air supply, per cent	10	50	75	100	150	200
Assumed temperature of escaping gases, deg. fahr.	400	450	475	490	Over 500	Over 500
Corresponding ideal efficiency of boiler, per cent	84.2	80.27	77.66	75.22	Under 70.94	Under 67.09
Possible saving in fuel due to reduction of air supply to 10 per cent excess, expressed as per cent of oil actually burned under assumed conditions	0	4.67	7.78	10.68	Over 15.76	Over 20.32

The 3 per cent used for boiler radiation is subject to some variation, being greater in small boilers and less in large units. For medium units, 3 per cent is probably very close.

16 The stack temperatures for any particular type of boiler, for any given load and corresponding to any assumed per cent of excess air, will vary with the size of boiler, arrangement of heating surface, character of baffling, condition of heating surface, etc. Stack temperatures will also vary with the different types of boilers corresponding to these factors. The temperatures given corresponding to the stated air supply, from 10 to 100 per cent excess, are those to be expected in ordinary practice and necessarily approximate: with boilers having three passes of gases and sinuous headers, the temperatures in general will be lower than those indicated; with boilers having but one pass and flow of gases parallel to tubes, the temperatures in general will be higher than those indicated.

17 Very few data are available for the temperatures corresponding to 150 and 200 per cent excess air, and the corresponding figures are given merely to show in a general way the magnitude of the losses easily resulting from careless firing of crude oil. The flue temperatures assumed are also subject to variation dependent on the rate of forcing the boiler and other well known elementary factors. The excess air with careless oil burning is usually greater than with careless coal firing, because of the greater excess draft power of chimneys.

In the preceding table the writer has calculated the saving that could be effected by reducing the air supply from that specified to an ideal condition assumed to correspond to 10 per cent excess air. This saving in fuel is of vast importance, but has been almost completely neglected with oil fuel.

18 It is possible to obtain a fair notion of the percentage of excess air from a mere determination of the amount of CO_2,—that is, assuming all hydrogen having been burned to H_2O and all carbon to CO_2. Any simple formula involving the element CO_2 must be dependent on an assumed percentage of hydrogen in the oil fuel, but inasmuch as the hydrogen contained is fairly uniform for any given grade of oil, there is but little error in such an assumption.

TABLE 3 POUNDS OF AIR PER POUND OF OIL AND RATIO OF AIR SUPPLIED TO THAT CHEMICALLY REQUIRED

Per cent CO_2 by volume as shown by analysis dry chimney gases	Light oil C, 84 per cent; H, 13; S, 0.8; N, 0.2; O, 1; H_2O, 1		Medium oil C, 85 per cent; H, 12; S, 0.8; N, 0.2; O, 1; H_2O, 1		Heavy oil C, 86 per cent; H, 11; S, 0.8; N, 0.2; O, 1; H_2O, 1	
	Pounds of air per pound oil	Ratio air supply to chemical requirements	Pounds of air per pound oil	Ratio air supply to chemical requirements	Pounds of air per pound oil	Ratio air supply to chemical requirements
4	51.40	3.607	51.93	3.704	52.45	3.803
5	41.31	2.899	41.71	2.975	42.12	3.054
6	34.58	2.427	34.90	2.490	35.23	2.554
7	29.77	2.089	30.04	2.143	30.31	2.198
8	26.17	1.836	26.39	1.883	26.62	1.930
9	23.37	1.640	23.56	1.680	23.75	1.722
10	21.12	1.482	21.29	1.518	21.45	1.555
11	19.83	1.391	19.43	1.386	19.58	1.419
12	17.76	1.246	17.88	1.276	18.01	1.306
13	16.46	1.155	16.57	1.182	16.69	1.210
14	15.36	1.078	15.45	1.102	15.55	1.127
15	14.39	1.010	14.48	1.033	14.57	1.056

19 Table 3 shows the calculated weight of air per pound of oil and the ratio of actual air supply to chemical requirements, for the various grades of oil and various percentages of CO_2. Under the present systems of firing the amount of CO_2 present in the flue gases is often as low as 4 or 5 per cent. With an ample supply of labor and a careful and scientific adjustment of dampers by hand, the percentage of CO_2 under an ideal and uniform load can be maintained as high as 13 per cent. With automatic control and under variable load conditions it has been found possible to maintain a high percentage of CO_2 conforming very closely to ideal conditions.

20 The first notable step in advance of the crude systems of hand firing was at the plant of the Pacific Electric Railway Company, Los Angeles, Cal., under the direction of J. R. Atchison, then Chief Engineer for that company, now Superintendent of Construction for Chas. C. Moore and Company, Engineers. Mr. Atchison developed a plan for firing 18 boilers, averaging 3 burners per boiler, totaling about 54 burners, with central hand control of oil pressure.

21 The operator was stationed near the oil pumps, which were run at a practically constant speed. In front of the operator were the oil pressure gage connecting to oil main and the steam pressure gage connecting to steam main. The operator's duty was to maintain a uniform steam pressure on the boilers by opening or closing a bleeder valve on the oil pump discharge line, thus increasing or decreasing the pressure in the oil main, and simultaneously the rate of firing of all of the boilers. The operating crew of the boiler room for each shift consisted of the one operator controlling the oil pressure and one water tender; which is probably the record to date for the minimum of boiler room labor for any plant of this size. It was a simple matter to substitute automatic regulation for hand control, following which the writer conceived the idea of utilizing this variation in oil pressure as a secondary means for controlling the supply of steam to burners and the air supply for combustion.

22 The writer will now explain the principle of operation, details of construction and results in actual trial of the Moore-Patent automatic fuel oil regulating system. This system controls the supply of oil to all burners, the supply of the atomizing agent to all burners, and the supply of air for combustion, for any number of boilers, all from a central point. The results are: increased boiler plant efficiency, the practical prevention of smoke, and decrease in the maintenance cost of boiler equipment, due to a more uniform manner of firing.

23 In this system all individual burner valves, both steam and oil, are opened wide or nearly so, and all burners are operated under full pressure in the respective mains. In the larger plants all dampers are connected to a common rock shaft and move simultaneously.

24 A slight variation in the steam pressure on the boilers, due to any variation in the demand for steam, is the primary means of control for a steam regulator or governor which varies the oil pressure at the oil pumps and in the oil main. Corresponding to an increased pressure in the oil main, there is an increase in the amount of oil fired and a rise in boiler steam pressure; and corresponding to a decrease of pressure in the oil main, there is a decrease in the rate of oil fired

and a lowering of the boiler steam pressure; this regulator thus maintains a uniform steam pressure on boilers at all loads within the governing limits. The variation in pressure in the oil main is the secondary means for controlling the supply of steam for atomizing purposes and also for controlling the supply of air for combustion.

25 The supply of steam to burners is controlled by regulating the pressure in a separate low-pressure main common to all burners, the pressure in this main bearing a certain predetermined relationship to the pressure in the oil main and being controlled by a ratio regulator. By means of a specially constructed diaphragm regulator, the opening of the boiler dampers is made to increase or decrease with a corresponding variation of pressure in the oil main, the change in damper opening, in turn, governing the supply of air for combustion.

MAINTENANCE OF A UNIFORM STEAM PRESSURE BY REGULATING THE OIL PRESSURE

26 Any reliable pump governor can be used to control the oil pressure in the oil main so as to maintain a uniform steam pressure. This governor can be used to operate either a throttle in the steam supply to the steam-operated oil pump, or a bleeder valve in the oil pump discharge pipe. The writer's experience has been most satisfactory in the use of the well known Spencer damper regulator, the movement of the power lever being used to control the bleeder valve in the oil pipe discharge, in lieu of connecting it to the damper, as is done in damper regulation service for coal burning plants.

27 Difficulty will be experienced in any endeavor to maintain a uniform steam pressure by regulating the oil pressure if it is attempted to effect this by controlling the supply of steam to the throttle of the oil pump. This is because of the surging of the oil pressure due to the alternate speeding up and slowing down of the pump; a difficulty that can be overcome in a measure by very careful adjustment, though the writer does not deem the attempt worth while. In such an attempt difficulties will also be encountered, due to the storage of heat in the boiler and boiler setting, and the time interval, corresponding to a change in the rate of firing the boiler, necessary to restore the steam pressure to the normal. The same difficulty will exist in a measure with regulation of the oil pressure by a bleeder valve on the oil discharge, provided it is attempted to connect the regulator so as to control the boiler plant automatically through the entire range of load.

28 By reason of the influence on the economy of plant of the rate of variation in oil pressure, and consequently in the intensity of fires, it is important that the ideal conditions be approached as nearly as may be; and to this end it has been found more economical to maintain, as nearly as possible, a uniform rate of firing with gradual changes in rate of firing due to changes in load on the boiler plant. When this system was first tried out at the Redondo Plant of the Pacific Light and Power Company, where the tests forming the basis of another paper presented by the writer at this meeting were made, the steam pressure regulator was used to control the entire plant of eighteen 600-h.p. boilers, although the steam-to-burner regulator and damper controller were connected only to the six boilers included in the No. 2 plant unit.

29 Before the installation of the automatic steam pressure regulator, central hand regulation was employed. Fig. 1 shows the record from a Bristol recording gage connected to the oil pressure main, and the corresponding record of the steam pressure, with hand regulation.

30 In adjusting the regulator for maintaining a constant steam pressure it was at first attempted to control the oil pressure for the entire range of load, and results somewhat similar to those indicated above for hand regulation were secured. It was finally found desirable to limit the regulation to such amount of opening or closing of the bleeder valve as would vary the oil pressure through a range of from one-fourth to one-third of the total variation required. When so operated the supply of steam to the oil pressure pump was direct from the auxiliary steam main and the throttle was hand-adjusted from time to time, whenever the range in load exceeded the limits of the steam regulator. Thus connected, the steam governor became an aid to the fireman, though not completely automatic in its action. When approaching either the upper or lower limit of the regulator, the condition is easily apparent from the position of the regulator yard arm, and the fireman in charge then slightly increases or decreases the rate of oil pumping, dependent on plant requirements. No additional labor is necessary on account of such connection, and except when passing over peak loads, it is unnecessary to adjust the oil pump throttle more frequently than every two or three hours. The corresponding oil and steam pressure records from such regulation are reproduced in Fig. 2, in which the curves are smoother than with hand adjustment for the oil pressure as well as the boiler steam pressure. These curves were obtained in a violently fluctuat ng

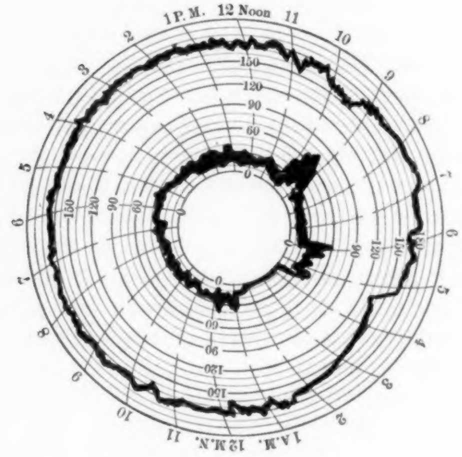


FIG. 1 RECORD SHOWING PRESSURES IN MAIN STEAM PIPE AND OIL-PRESSURE MAIN, CORRESPONDING TO HAND REGULATION OF OIL PRESSURE INNER CURVE OIL PRESSURE, OUTER CURVE STEAM PRESSURE.

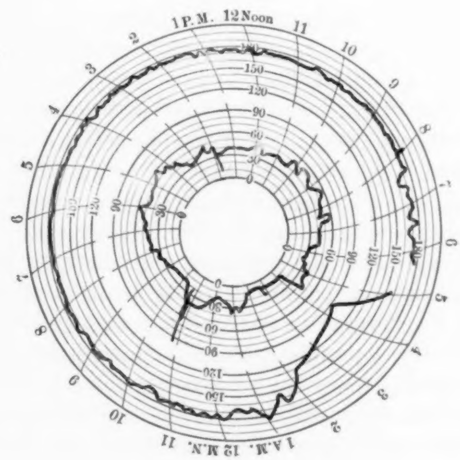


FIG. 2 RECORD SHOWNG PRESSURES IN MAIN STEAM PIPE AND OIL-PRESSURE MAIN, CORRESPONDING TO AUTOMATIC CONTROL. INNER CURVE OIL PRESSURE, OUTER CURVE STEAM PRESSURE.

railway load, where the momentary swing in load would vary at times from 20 to 100 per cent of the unit capacity.

31 The increased economy due to such uniform rate of firing is self-evident, as well as the saving in the attention required for the adjustment of the steam supply to burners and of the air supply for combustion. In actual experience this system of regulation has been found absolutely reliable and the firemen have been found not only very much interested in it, but exceedingly anxious for its installation, owing to the saving in labor and the superior steam and oil regulation produced.

STEAM-TO-BURNER REGULATOR

32 Before the development of this regulator, tests were made to determine the relationship between the amount of steam required for atomizing oil and the amount of oil burned. Pressure gages were connected on the branch pipes for both oil and steam, between the respective throttle valves and the burner head. The burner was fired at various measured rates and record made of the steam and oil pressures. It was found that for a variety of burners this relationship could be represented by a straight line, the required steam pressure being equivalent to the product of the oil pressure by a constant, plus a fixed pressure difference. With the outside mixing burner used at Redondo, this relationship was about as follows: Steam pressure = 3 × oil pressure + 20. The constants for ratio and difference will vary with the type of burner and the relative sizes of burner orifices for steam and oil, as well as with the viscosity and temperature of the oil.

33 With this fact established, it became a simple matter to design a regulator, of which the essentials were opposing steam and oil diaphragms with areas and leverages conforming to the observed ratio, and a weight element equivalent to the required difference in pressures. Accordingly a simple regulator was designed, the final development of which is as shown in Fig. 3. In this regulator the upward pressure is exerted on two diaphragms, the one at the left subject to the oil pressure in the oil main, the one at the right to the steam pressure in the low pressure steam main connected to the burners.

34 The fulcrum is adjustable for any desired ratio of leverages. The yard arm is counterbalanced at the left, and weighted at the right for the desired weight constant due to the pressure difference factor. Whenever equilibrium is disturbed by variation in oil pressure, the

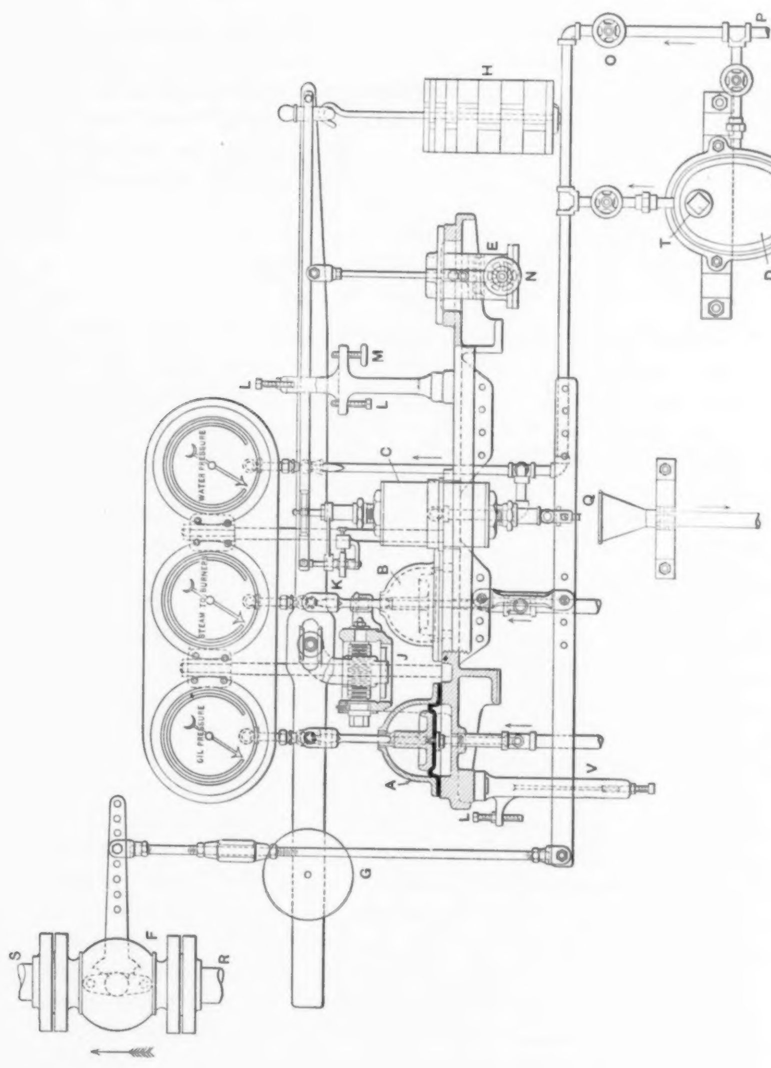


FIG. 3 STEAM-TO-BURNER REGULATOR

main yard arm will be compelled to move either up or down. This motion is communicated to a water cylinder or motor, such as is used on the well known Spencer damper regulator. The movement of the water cylinder in turn operates a suitable lever and connecting rod and finally a rotary chronometer valve, which increases or decreases the supply of steam to the low pressure steam main in such a manner as to restore equilibrium, thereby providing the desired increase or decrease corresponding to the initial change in oil pressure. The pressure gages shown indicate at a glance the pressure on the oil main, the pressure on steam-to-burner main and the water supply pressure actuating the water cylinder. The following is a list of the lettered parts in Fig. 3:

A Diaphragm actuated by pressure in oil-to-burner main
B Diaphragm actuated by pressure in steam-to-burner main
C Water motor
D Water strainer
E Dash-pot
F Conical seated chronometer valve
G Sliding counter balance
H Adjustable weights
J Fulcrum adjustment
O By-pass valve around water strainer
P Water pressure inlet
R High pressure steam from boilers
S Low pressure steam to steam-to-burner main
V Guide for lever with stop screws

35 The writer would refer those not familiar with the operation of the Spencer water cylinder to a descriptive catalogue of the Spencer damper controller. This regulator has been entirely successful in actual experience, and is completely automatic in action; although on swinging loads a further saving in steam supply to burners can be effected by a slight adjustment of the amount of weight on the yard arm. This adjustment will cause a slight constant increase or decrease in the steam-to-burner pressure main, as observation of fires dictates.

36 When boiler fires are started, it is usually a fact that the oil in the main is a little colder than under normal conditions, corresponding to which a greater oil pressure is required; this increased oil pressure in turn provides the necessary increased steam pressure and supply of steam for the atomization of the colder oil. There is a further furnace effect, as not only is an increased amount of air for combustion required when furnaces are cold, but also a greater

amount of steam for atomization. Correspondingly, after boilers have been fired for a number of hours, or on cutting down the load after having passed over a peak, the furnace walls will be found somewhat hotter than the normal, corresponding to which a slight reduction in the amount of steam for atomization is permissible, as well as in the air supply for combustion. A slight adjustment of the weights at such critical periods of the load will give a slightly increased economy.

37 If the grade of oil varies from day to day, and if the temperature of oil in the oil main is not maintained uniform, this condition can also be corrected by weight adjustment. It is essential that the oil main be of a size that will make the pressure substantially uniform throughout the entire plant, otherwise there will be a variation in the amount of oil fired by the individual burners. It is also essential that the low pressure steam main be of such size that the steam pressure will likewise be uniform throughout the plant.

38 In addition to the economy effected by this system due to a saving in steam-to-burners, there is a secondary economy in the more efficient combustion of oil due to proper atomization at all times.

DAMPER CONTROLLER

39 In the development of the damper controller, tests were made to determine the relationship between variations in the amount of damper opening and variations in the pressure of oil on the burner orifice. It was found that this relationship could be represented approximately by a straight line, although the total amount of damper opening was not found to be directly proportional to the total oil pressure.

40 In the development of this regulator the writer regarded perfect regulation as an impossibility, and any degree of regulation giving even approximately the proper amount of air for variations in loads, as an important step in the economical operation of oil burning plants. Accordingly a trial regulator was constructed, having a diaphragm actuated by the pressure of oil in the oil main, the movement of the diaphragm being opposed by a lever connected to a coil spring. In this regulator the amount of movement of the diaphragm and of the lever was proportional to the oil pressure after passing the point of zero spring tension. This amount of motion of the main lever was multiplied and transmitted to a rock shaft operating the six boiler dampers in No. 2 unit, by means of a water

controlling valve and a suitable hydraulic cylinder connected to the rock shaft.

41 As a result of the experiment it was found that by a cut and try process, with levers and rods at various angles, a movement of dampers could be secured through the working range of load, giving an air supply at all loads but slightly in excess of theoretical requirements.

42 The final development of this regulator for large plants is shown in Fig. 4. In this regulator the large diaphragm is subject to the upward pressure corresponding to that in the oil main. The motion of the lever is opposed by a coil spring adjustable in position along the lever so as to obtain the required range of motion of the dampers corresponding to the given range of oil pressure. By means of a double-ported controlling valve, connected to the damper controller, the supply of water under pressure is admitted either to the top or the bottom of the hydraulic cylinder connected to the rock shaft. The control is on the principle of the well known differential lever, such as is used in a steering gear, and also common to many types of damper regulators.

43 After being once properly adjusted for its entire range of motion, this regulator gives entire satisfaction. It is subject to variation from certain external influences, such as the temperature of the boiler furnace on approaching or receding from peak loads or in starting fires, and variations in the temperature and density of oil. Immediate adjustment and correction for these difficulties can be made by a slight turning of the hand wheel on the coil spring, giving a constant change in the amount of opening or closing of all dampers. In actual experience, no adjustment of the damper controller has been necessary except at intervals of three or four hours, depending on the nature of the load carried, the frequency of peaks, etc.

GENERAL ARRANGEMENT OF REGULATORS AND PIPING

44 Fig. 5 and 6 show the arrangement of boiler plant and the main piping connections for the steam-to-burner regulator and damper controller. The writer has deemed it unnecessary to illustrate the oil pump governor for maintaining a uniform steam pressure as it involves nothing of novelty in its construction.

45 In order to prevent the corrosive action of the oil fuel on the rubber diaphragms it has been found necessary to interpose a water cylinder in the oil pressure line, as shown in the illustration. A pipe

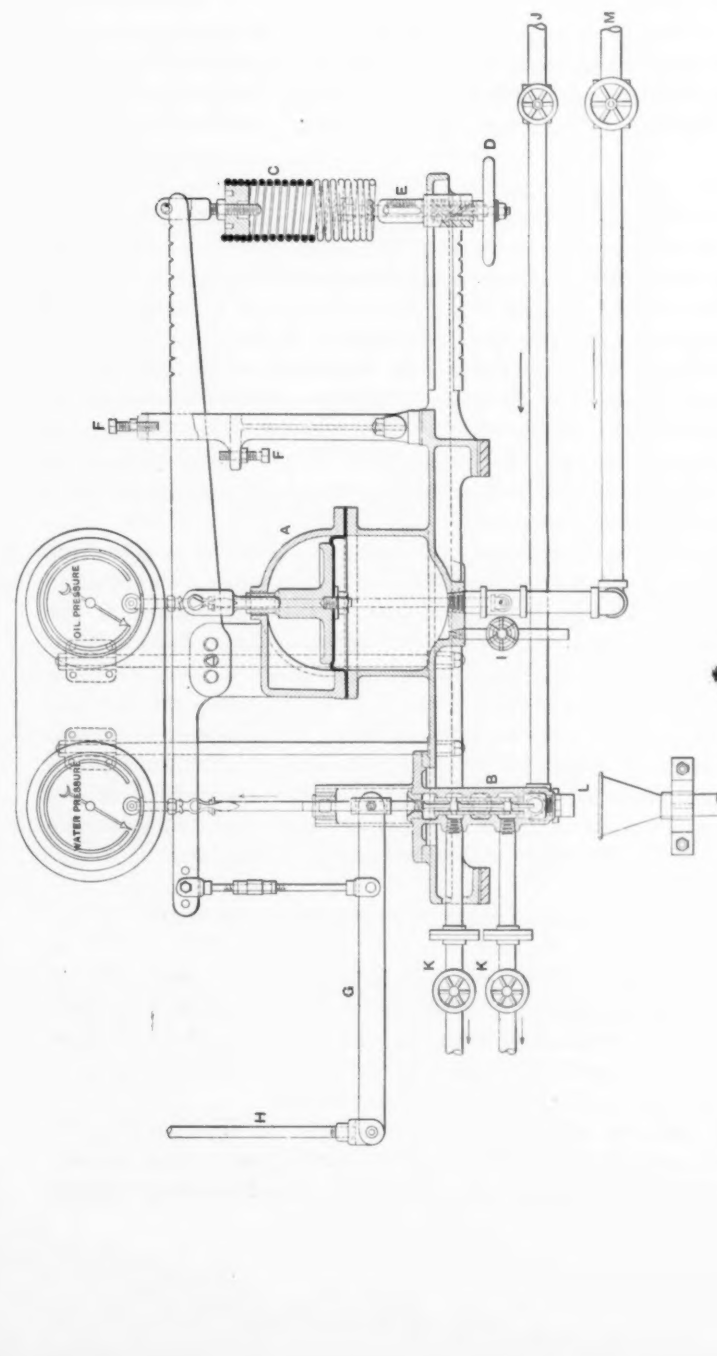


FIG. 4 DAMPER CONTROLLER

A DIAPHRAGM ACTUATED BY PRESSURE IN OIL-TO-BURNER MAIN
B WATER CONTROL VALVE OPERATING HYDRAULIC RAM
C ADJUSTABLE COIL SPRING
G DIFFERENTIAL LEVER
H CONNECTING ROD FROM ROCK SHAFT TO DIFFERENTIAL LEVER
J WATER PRESSURE INLET TO CONTROL VALVE
K SUPPLY AND RETURNING LINES TO HYDRAULIC CYLINDER
M PIPE CONNECTION TO INTERMEDIATE WATER RESERVOIR

from the oil pressure main is connected to the top of this cylinder. Water connection is provided from the bottom of the cylinder to the steam-to-burner regulator and to the damper controller, thus protecting the diaphragms from oil. This water cylinder is connected with the city main for convenience in filling necessitated by leakage, etc. The chronometer valve on the steam-to-burner regulator must be carefully proportioned for the maximum requirements of the plant, and the relative amount of opening due to the regulator so adjusted. A pipe stanchion should connect the steam pipe to the floor to prevent change in its position relative to the chronometer valve. The chronometer valve should be by-passed so that the supply of steam to the burners can be regulated by the hand throttle valve shown, when it is necessary to cut out the regulator and to clean the pilot valve of the Spencer water cylinder; the frequency of this operation is dependent on the hardness of the water supplied under pressure for operating the damper.

46 The installation of the damper controller is simple, but it is important to place the controlling lever connecting from the rock shaft to the differential lever at such an angle as to give the right amount of damper opening under all conditions of load.

47 The piping to the hydraulic cylinder should be so valved that when it is necessary to cut the damper controller out of service for replacement of the diaphragm, etc., the movement of the hydraulic cylinder can be effected by hand control of the valves in the pipes supplying the pressure water to the opposite ends of the hydraulic cylinder.

48 The rock shaft should be counterweighted, so that in case of accident the counterweights will open all dampers. The connection from each individual damper to the rock shaft is by means of a chain, and the dampers are also supplied with individual counterweights and so arranged that any one may be adjusted by hand when conditions require it. It is desirable to limit the minimum and maximum opening of the dampers by means of suitable collars on the plunger rod of the hydraulic cylinder. The minimum opening is determined by the extent to which the dampers can be closed without causing gas to escape into the boiler room, or without producing explosions of the gases which collect in the furnaces and gas passages, and the maximum is determined by the plant requirements.

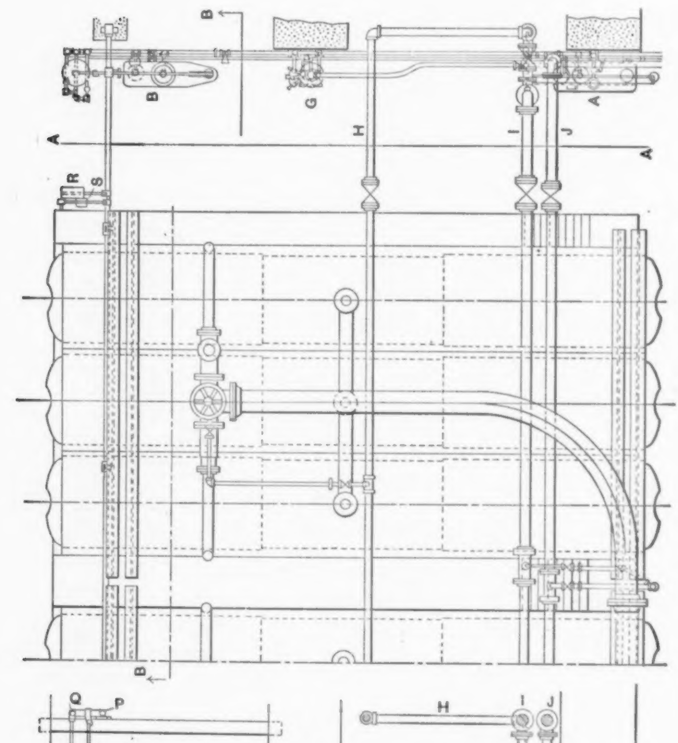
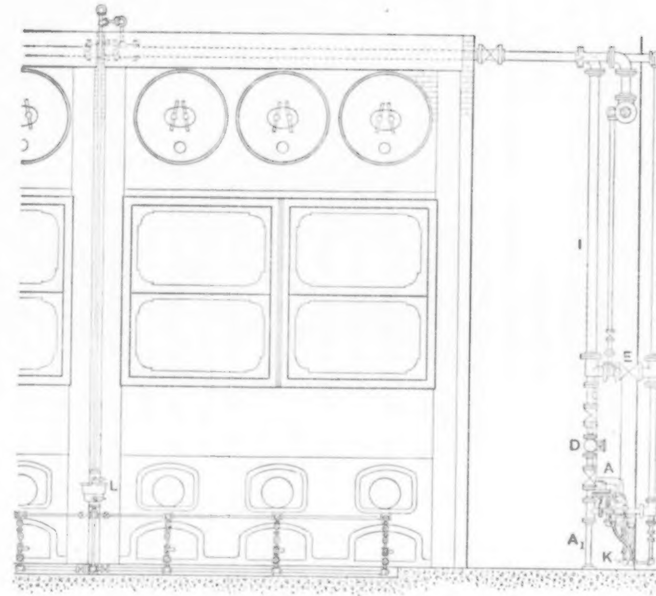


FIG. 5 GENERAL ARRANGEMENT OF APPARATUS—SECTIONAL ELEVATION AND PLAN

LIST OF PARTS, FIG. 5 AND 6

- A Steam-to-Burner Regulator
- B Damper Controller
- C Hydraulic Ram Actuated by Damper Controller
- D Chronometer Valve Actuated by Steam-to-Burner Regulator
- G Water Chamber, Subject to Oil Pressure, to Protect Rubber Diaphragms
- H High Pressure Steam From Boilers
- I Low Pressure Steam to Steam-to-Burner Main
- J Oil-to-Burner Main
- K City Water Pressure Main
- L Oil Strainers
- P Damper Rock-Shaft, Operated by Ram
- R Balance Weight
- V Connecting Rod, Rock-Shaft to Differential Lever



FRONT ELEVATION

Fig. 6 General Arrangement of Apparatus—Front Elevation and List of Parts

OPERATION OF THE COMPLETE SYSTEM

49 It is unnecessary to disconnect any of the regulators during the starting or stopping of the plant, as these will perform their respective functions automatically.

50 In case of accident, the steam pressure governor can be cut out by merely closing a valve between the oil main and the bleeder valve; when the oil pressure can be varied by hand control of the pump throttle. To cut out the steam-to-burner regulator, it is necessary merely to open the by-pass valve and close the other two valves (shown in Fig. 3). To cut out the damper controller, it is necessary only to close a valve in the supply pipe to the water controlling valve and to open to waste the valves connecting with the water cylinders, when the counterweights will immediately throw all the dampers wide open and permit individual hand control. The hydraulic cylinder can be used to control all the dampers as explained above by controlling by hand the supply of water under pressure to power cylinder. Any regulator can be eliminated without interfering with the functions of the remaining regulators, and any boiler may be cut out of service without affecting the regulators.

51 After the fires are started as usual and the boilers are fairly well under load, it is customary to equalize all the fires and to make a refined setting of both steam-to-burner regulator and damper controller. The respective dampers having once been properly set, all the fires may be equalized by a slight closing or opening of all the oil burner valves, so gaging the supply of oil as to produce the desired excess of air, which condition will be known to exist when the right degree of smoky haze appears in the gases well beyond the combustion zone, as viewed through peep holes provided for the purpose. The next operation is to equalize the supply of steam-to-burners by slightly opening or closing all steam supply valves on the burners. If a greater damper opening is required for the given load, adjustment of the spring tension will give the desired results. If a change in the total supply of steam-to-burners is necessary, a weight adjustment at the regulator will suffice. Thereafter with the same character of fuel oil, and with care in the maintenance of a fairly constant temperature of oil, no further adjustment of regulators or burners will be necessary until a peak load has been passed, when a further slight adjustment should be made for reasons explained above. This individual adjustment of burners is necessary owing to the fact that the burner orifices become unequally worn and fouled by use.

52 While this system results in an appreciable saving in manual labor, it does not dispense with any considerable number of firemen. The firemen should observe the working of fires continually to make certain that exterior causes are not preventing the regulators from performing their respective functions, and it is only by careful and continual observation that the firemen are enabled to maintain the highest economy, by correcting the external conditions which the regulators are incapable of anticipating, with the refined adjustments mentioned above.

53 Contrary to the usual attitude of power plant operatives toward automatic contrivances, it is a fact that all firemen who have had experience with these regulators so far willingly observe instructions regarding their use and coöperate to secure favorable results. It is only natural that firemen should regard this system with favor, inasmuch as the manual labor of attending to boilers is greatly lessened. The time made available should be spent, however, in giving more attention to the refined adjustments important to securing the highest economy.

54 On momentary changes of load of considerable magnitude, sufficient to cause appreciable change in the boiler steam pressure, all the regulators will act promptly and synchronously. The writer has observed this apparatus during periods of short circuits on a station, followed by an interval during which no power was developed. All the regulators performed their functions admirably during this change, from maximum to minimum position, and during the succeeding opening up of dampers and building up of fires to normal conditions.

55 After a certain set of conditions become established for a given load, including the rate of flow of chimney gases, temperature of gases, the draft, etc., the regulator will respond more readily than the chimney to momentary changes. Owing to the fact that the damper controller is set for the continuous operating conditions at each load, it frequently happens that the chimney does not respond with sufficient rapidity, causing momentary smoking. If the load remains unchanged for a short period then normal conditions will gradually establish themselves, and smoking cease, except for a slight haze required for economical oil firing.

56 It is not usual that reliability of operation results from the use of automatic contrivances and the trend of engineering is adverse to their use. In actual practice it has been shown, however, that these regulators are completely dependable, a case of failure in actual

service not having resulted since the completion of this system at the Redondo Plant. It will be noted that the regulators are massive in design and simplified to the extreme. There are certain parts of this system which require care, and when such care is bestowed there is little liability of any interruption in service. This system requires a continuous supply of water under pressure which is ordinarily obtained from the city mains. The writer recommends a reserve supply tank on the power plant roof, and also a connection, to be used when required, from the boiler feed main.

57 Records in coal burning plants showing failure during operation of the better types of ordinary damper regulators, are certainly few. There is possibly no single automatic apparatus more compliplicated than a modern engine shaft governor with its valve gearing, particularly on a large engine, but its success is due to its massive design and its power to do work in comparison with the opposing forces; and it is universally accepted without question as to its reliability.

58 The writer regrets his inability to present a statement showing the saving in fuel effected by this system, as compared with hand operation. In tests made at the Redondo Plant, a saving is apparent, but the figures are too involved for convenient presentation. As compared with other units of this plant not so regulated, the evidence of lower temperature of flue gases, and more even regulation, are significant. The saving to be expected from the use of this system would naturally be greatest in plants where the load is most widely variable; as with such a load, it is impossible, with hand adjustment of dampers, to work sufficiently close to the economical requirements of air supply.

59 In small stations, the pressure governor and steam-to-burner regulator would be of a design similar to that illustrated. The damper controller, however, would be simplified by the omission of the separate hydraulic power cylinder, there being used an apparatus similar to the Spencer damper regulator, a tension spring being substituted for the weight on the yard arm, commonly used in coal burning practice.

60 The writer would urge central regulation of oil pressure regardless of the size of the plant. It would be possible in large stations to have central control of the steam supply to burners, and of the dampers. It will usually prove more convenient however, by reason of possible variation in the oil pressure in different parts of

the plant, to use separate steam-to-burner regulators and separate damper controllers, on each unit or panel, up to the limit of about 4000 or 5000 h.p.

This system is covered by letters patent.

CONCLUSION

61 The necessity and importance of careful regulation of air supply, in connection with coal burning plants, has of late received such widespread attention, as to make self-evident the importance of some system of regulation in connection with oil burning stations. Such a system as herein described is a distinct advance in the science of burning crude oil, and is worthy of the careful investigation of those interested in the economical production of power in plants using crude oil. It is believed that a modification of this system would yield readily to the requirements of boiler plants using natural gas as fuel.

DISCUSSION ON THE TWO PRECEDING PAPERS

MR. GEO. H. BARRUS The author states in the first paragraph that this plant gave a "notable economy," which means, I suppose, that it gave a better economy than other large electric steam power plants. It would add much to the value of the paper if he would present examples of what has been accomplished by other plants, so that we could see to what extent the economy shown is really notable.

2 It would also be interesting, and of great value, if he would tell us something about the economy of the individual elements of the plant, i.e., what was the efficiency of the boilers taken by themselves; also what was the steam consumption of the engines per i.h.p. per hour. With information of this character, the members of the Society would be able to form some idea themselves as to whether the plant was economical. Upon this subject there is no intimation in the paper as presented.

3 It seems to me highly improbable, in connection with a test of such a character as the one described, involving, as the mechanical papers have told us, such a large pecuniary bonus, and one which required the assistance of such a large corps of testing men as the number referred to, that some one did not make an evaporative test of the boilers, and steam consumption test of the engines, to determine their individual

economy. Very likely the author has such data, and I will be glad to hear from him on that point.

Prof. William Kent The papers are very interesting as showing what can be done with oil and with reciprocating engines on the Pacific coast. The statement in Par. 68 of the paper on Fuel Economy Tests, that "Inasmuch as the steam turbine gains largely in apparent economy due to increase in vacuum, similar reductions should be made in the stated economy of such plants, and correspondingly in the reported fuel consumption," indicates that the author considers the result of the test a proof that reciprocating engines are as good as turbine engines; and seems to be equivalent to saying it is believed that the steam turbine is good because it can utilize a high vacuum, which the reciprocating engines could not do economically: and that for this reason we must deduct from the economy of the turbine the amount it gets from the high vacuum and charge that against it. I do not think that is fair to the turbine.

2 These tests show very good results in kilowatt-hours per barrel of oil. Between the barrel of oil and the kilowatt-hours there are many variable conditions. First, there is the oil-burner. I understand that they have in San Francisco a new burner which is more efficient, especially in the regulation of the air supply. It appears that part of this higher economy in the San Francisco test is due, not to extraordinary efficiency in heating surface of boilers, but to the regulation of the air supply. Second, the efficiency of the boilers in San Francisco is due to the fact that it is theoretically possible to get a greater percentage of efficiency out of oil than out of coal. I know of no large plant in the East tested with oil. We have to test with coal, where we are troubled with ashes, the large excess of air necessary to burn coal properly, and many disadvantages which the writer would credit to the coal pile, because it suffers these disadvantages.

3 Thus we have in favor of oil, the theoretical economy higher than that of coal and the diminution of the excessive losses due to unregulated air supply in coal firing. The records of the test fail to show, however, how much of the total economy between kilowatt-hours and barrels of oil is due to the reciprocating engine. As far as I can find, not a single point is made in favor of the reciprocating engine, other than the general statement that because we have the great economy the reciprocating engine is a good engine.

4 In making a test between the reciprocating engine with oil as fuel

under the boilers, and a turbine engine with coal, we have so many variables that from such a test-record a statement of the relative value of the reciprocating engine and the turbine is impossible. The only way to get such a statement is to have a test made with the turbine in San Francisco with oil, with the same boilers and with same adjustment of air supply, and Professor Jacobus to watch it, and we will get as good results out of the turbine, if not better, than have been obtained from the reciprocating engine.

Prof. William D. Ennis It is difficult to say whether in reality this plant is showing notable economy. A heat consumption of 25 288 B.t.u. per kw-hr. is perhaps unprecedented, if it is maintained in the ordinary operation of the plant. This rate, noted during

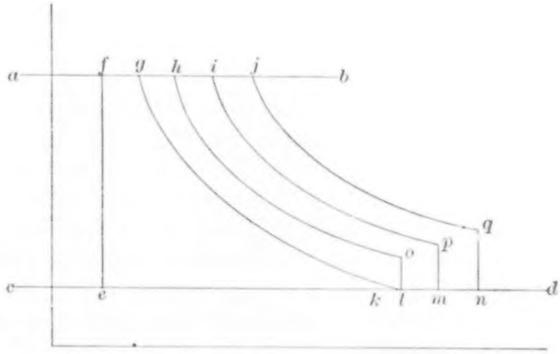


Fig. 1 Comparison of Rankine and Clausius Cycles

the 15-day test, under expert superintendence, is, however, not comparable with the 2 to 2½ lb. coal consumption rates of our best large plants day in and day out as the load factor comes. It is to be regretted that the details of the boiler performance are not given, as this would permit of judging as to the economy of the engines. Assuming that the boiler efficiency was 0.80, and the efficiency from engine cylinder to switchboard 0.855, then the heat unit consumption from line 24 of Table 1 was about 215 B.t.u. per i.h.p. per min.—not at all an exceptional result with superheat, under test conditions. With engines of this size, operating at 180 lb. pressure, and with certainly a fair vacuum—28.334 in.,—one would almost expect a better result. If these were service tests, however, then the results are of course most excellent. But the results do not seem to be those to be anticipated in regular running.

2 The proposal to compare engine performances with the limits possible in a standard cycle has been made before. There is no good reason for calling this standard cycle that of Rankine. It is true that Rankine recognized that the terminal pressure might be *equal to*, greater than or less than the condenser pressure (*The Steam Engine*, 1897, Art. 278); but all of his computations are based on the existence of some terminal drop. On the other hand, Clausius (*Fifth Memoir, On the Application of the Mechanical Theory of Heat to the Steam Engine*) clearly describes what I suppose we have all had in view as the standard cycle; viz, one in which adiabatic expansion is followed by isothermal condensation without intermediate drop in pressure. The expression for efficiency of this cycle is perfectly definite, based on the upper and lower temperatures: that for the Rankine cycle is not. The illustration may make this clear. The Rankine cycle between the limits *a b* and *c d* might be any one of *fjqne*, *fipme* or *fhole;* the Clausius cycle can be only *fgke*.

MR. J. R. BIBBINS Much information of an important character has been omitted from the valuable paper on Fuel Economy Tests; namely, data on individual efficiencies of engine and boiler plants respectively. It may have been difficult to obtain such data during the test, but they are certainly essential to the conclusion reached by the author, whereby the reciprocating engine is credited, by inference if not directly, with the major part of the excellent results obtained although no data are presented which give the least possible ground for such a conclusion.

2 Suppose, however, we take the author at his word and compute the efficiency possible from a steam motor working under the precise conditions stated. Referring to the 15-day test, the average load was about 3660 kw., and it is stated that the engines were rated at 4000 kw. Presumably, therefore, the engines during this 15-day test were operated at approximately their point of best economy. Now on the other hand, consider a turbine plant similarly rated at or near its point of best economy (just before the overload valve comes into service). If we supply this turbine with steam at 180 lb. pressure, 82 deg. superheat and a vacuum of $28\frac{1}{3}$ in., or even 28 (allow for drop between condenser and turbine), it is a question whether the author would place his opinion on record that the economy thus obtained in connection with the same oil-burning boiler plant would be inferior to the corresponding economy obtained by test on the reciprocating engines. In other words, with a given boiler plant

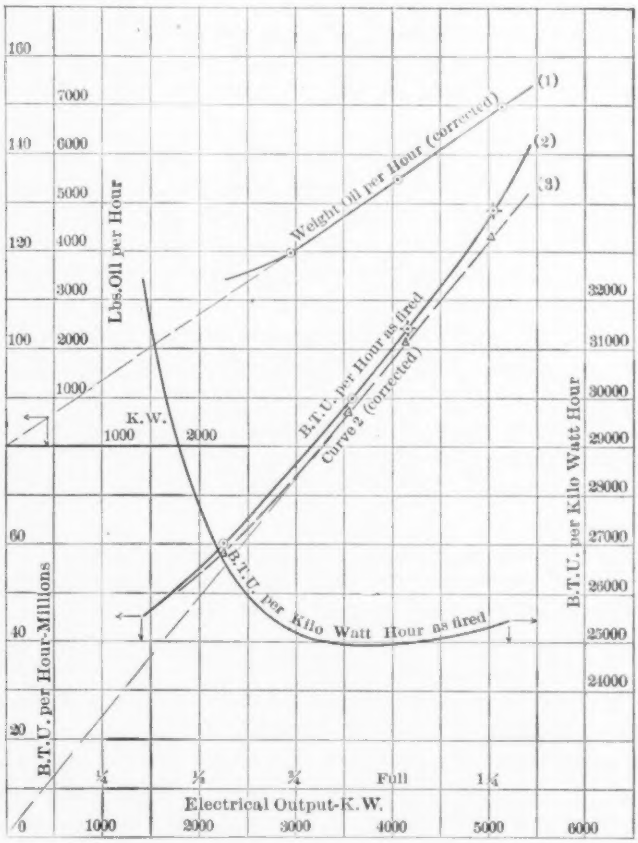
and given conditions of operation, does the author claim for the reciprocating engine economy superior to that which recent tests of large turbines have demonstrated beyond reasonable doubt? This is to be inferred from his closure. In Par. 68, the author states that, "inasmuch as the steam turbine gains largely in apparent economy due to increased vacuum," the reported economy from turbine plants should be reduced accordingly for comparison with engine-driven plants. The vacuum reported in the Redondo tests is fully as high as is obtainable in the best turbine plants in the country—28 and 28.4 in., therefore the necessity for correction is not apparent.

3 The crux of this whole matter seems to be the efficiency of the boiler plant with oil fuel. Three years ago the United States Naval Bureau of Steam Engineering conducted extensive tests with both oil and coal fuel upon a marine water-tube boiler, with the specific purpose of ascertaining the comparative boiler efficiencies respectively obtainable. Detail figures are not at hand, but the average ratio of evaporation between coal and oil, under otherwise identical conditions, was in the neighborhood of 10 to 15; and under normal power-plant conditions of land practice, the ratio was reported still higher, 10 to 17. Correcting for the difference in heat-values of fuel burned, it is apparent that the efficiency of an oil-fired boiler plant is from 10 to 20 per cent higher than that of one burning coal.

4 Assuming then a turbine plant generating a kilowatt-hour on 15 lb. of steam, under the conditions obtaining during the Redondo test, and a boiler plant burning coal with the efficiency of an oil-fired plant, evaporation would be at least 10 lb. of water per pound of coal, and the overall fuel consumption 1.5 lb. per kilowatt-hour or 21 000 B.t.u. per kilowatt-hour, an economy superior to that obtained at Redondo. That the turbine economy above noted is quite conservative is apparent from the results of tests upon a large turbine at the Waterside Station of the New York Edison Company. An average economy better than 15 lb. was obtained under conditions of 175 lb. pressure, 96 deg. superheat and 27.3 in. vacuum. It does not seem, therefore, that the reciprocating engine has demonstrated itself in the Redondo tests capable of any extraordinary economy, *per se*.

5 The shape of the economy curve A, Fig. 3, seems to establish a rather novel load characteristic for a power-plant unit; viz. practically constant economy over an extraordinary range of load. If

we consider the reciprocal of this curve in pounds of oil per kilowatt-hour, and again convert this into total oil per hour plotted against load, we have a so-called Willans line, shown as curve 1 of the accompanying figure. This Willans law is highly valuable in analyzing the rational performance of both steam and gas plants, and at a glance indicates unusual results. In this case, the load oil-consump-



Load in Kilowatts per Pound of Oil and per B.t.u. per Hour

tion curve is practically a straight line from the origin above about 3000 kw., curving rapidly below this point, a phenomenon for which it is difficult to find a good explanation. If now we plot a similar Willans line for load B.t.u. per hour, taking the observed values (line 24, Table 1), curve 2 is obtained. This line of total heat con-

sumption appears entirely rational as a characteristic of a steam engine governing by cut-off. It shows a decided curvature, the relative heat consumption increasing on both heavy and light loads. The point of tangency of a radial line from the origin indicates the point of best economy to be in the neighborhood of 4000 kw., which is entirely rational.

6 Finally, if the values obtained by correcting oil consumption according to contract stipulation are plotted (line 23, Table 1), the Willans line flattens out into the straight line 3, above 3000-kw. load, but shows an even more marked curvature below this load. This would indicate a practically constant heat consumption of the unit from 3000 kw. up.

7 The reasons for these peculiar characteristics are not clear, and from the known characteristics of a reciprocating engine tested at different loads, as was done in this case, it may only be inferred that the reasons lie in a variable boiler-plant efficiency. To my knowledge there is no prime mover possessing a straight-line characteristic through the origin, i. e., an economy remaining constant with the change of load. Consequently, we may look to the boiler plant only for an explanation of these results. And it is to be hoped that the author may be able to submit further data. The Redondo tests are so thoroughly authenticated in every respect as to preclude the usual possibility of errors of observation, so that the opportunity is exceptionally good for throwing further light upon oil-fired boiler performance. This would be particularly desirable in connection with the discussion of the methods of firing in the author's second paper.

Mr. I. E. Moultrop. This is an interesting paper upon a subject which has been somewhat neglected by the Society in recent years, and in the writer's opinion more papers on similar lines would be of value. Mr. Weymouth has made a careful test of the Redondo plant, and is to be congratulated on having built a fine station and one which shows remarkably good economy.

2 In view of the care with which this test was made and the large number of trained observers employed, it is to be regretted that more information was not obtained at intermediate points in the cycle which would point out specifically how this fine economy was obtained. There is nothing to show whether it was obtained in the boiler plant, the engine plant, the condenser plant, or the general station design. The author apologizes for the vacuum and attributes the lack of a

better one to the temperature of the cooling-water, which he considers high. The vacuum seems to be remarkably good under the given conditions, however.

3 Mr. Weymouth is in error when he states that Eastern power stations obtain their higher vacuum only by the use of cooling-water at a temperature near the freezing point. The writer knows of a number of large power stations where the condensing apparatus is designed to give vacuums inside of 1½ in. absolute with the circulating water from 5 to 10 deg. higher than the temperature stated in the paper; and he is inclined to think that a material increase in the cooling surface of the condenser, other conditions being the same, would have considerably improved the vacuum in this instance.

4 Some of the conclusions drawn by the author seem unwarranted, especially where he unfavorably compares large turbine stations in the East to his station. There is nothing in the paper which justifies the assumption that the engineers did the best thing when they selected steam engines for the prime movers. As the superior economy of steam turbines over reciprocating engines as prime movers for electrical generating stations has been so well demonstrated, and as the vacuum carried on the plant during the test was not especially good, only the boiler plant is left to be considered in accounting for the good results obtained. Tests in other places, and notably by the United States Geological Survey, indicate that boilers fired with good fuel oil show a considerably higher economy than with first-class steam coal.

5 The central station manager naturally wants a generating plant which will show a fine performance on the B.t.u. basis, but he is much more concerned in having a generating station which will put the maximum of kilowatts on the switchboard for a dollar spent in operating; and while the geographical situation of the Redondo station was doubtless such that fuel oil was a wise choice, no cost-information is given, and there is therefore no means of determining whether or not an engineer in the middle or eastern states would be justified in building a station for the use of fuel oil.

6 There is one point in the operation of this station which the author has not touched, and which I hope he will include in his closure, and that is the matter of the successful removal of the cylinder lubrication from the condensed steam before it is returned to the boilers. This has been a very serious objection to the use of surface condensers with steam engines in power plants, and it would be extremely interesting to know, either how the oil is successfully removed or

what difficulties and extra expense are entailed in operating the plant by reason of more or less oil getting in the boilers.

MR. A. H. KRUESI The subject has been well covered by those who have already discussed the paper. The economy hinges on the efficiency of the boilers. It appears that the conclusions on the last page are hardly justified without more data as to where the economy of the plant is to be found. The boilermakers might claim a large part of the credit and as has been shown a large part of it is due to the use of a fuel which is little short of ideal. It can be controlled instantly, produces very little flue dust, and has many other advantages which need not be detailed here.

2 It may be noted that the conclusions as to the flexibility of these units make a virtue of a necessity of the steam engine. By the same argument it might be contended that it is desirable to use five 1000 kw. turbines, instead of one 5000 kw. turbine, to reduce the liability of breakdown and increase flexibility of operation.

MR. F. W. O'NEIL The paper upon Fuel Economy Tests is of value in showing the performance of engines at fractional loads. As pointed out in Par. 58 the losses are proportionately greater at fractional loads, yet as the difference in losses at full and fractional loads is relatively small, the economy of the engines fairly approximates the economy of the whole station.

2 The result obtained of 252.8 kw.-hr. per bbl. of oil, or 25 288 B.t.u. per kilowatt hour, is by no means extraordinary when compared with the performance of engines reported to the Society in the past and when the size of the plant, the high vacuum and superheat are taken into account, as well as the high boiler efficiency possible when firing with oil.

3 The thing which is extraordinary in this report is the low guarantee accepted by the purchaser of 170 kw.-hr. per bbl. of oil, especially when it is considered that a large bonus was to be paid for every kilowatt hour secured over and above this amount. The difference between the guaranteed economy and tbe economy obtained on test is 32.8 per cent, or, in other words, the plant used 32.8 per cent less fuel than would have been consumed had the actual economy corresponded to the guarantee. This fact, together with the conclusion of the writer, might lead some to believe that the difference between the guaranteed and obtained economy represented the accuracy with which designing engineers were able to predict engine performance.

The writer believes it desirable to call attention to the fact that the above difference is by no means a fair measure of the ability of steam engine designers to predict engine performance.

4 It is to be regretted that water was not weighed, or, if it was, that the amount was not included in the results, so that the separate performance of boilers, engines, etc., could be segregated.

PROF. W. B. GREGORY The economy shown in these tests is undoubtedly due to high boiler efficiency. Crude oil is used extensively in the southwest, especially in Louisiana and Texas, and on the Pacific coast. Boiler tests with various kinds of coal are numerous. The tests in which fuel oil was used are much more limited as some of the best work has been done by boiler manufacturers and is not available to the general public.

2 During the last seven years, the writer has made more than fifty tests with fuel oil and has had a widely varying experience with this fuel. The best results can be attained only when careful attention is given to every detail of the boiler equipment. Burners for fuel oil use from less than 3 per cent to 12 or 15 per cent of the total steam generated. The arrangement of the furnace has been found to affect results to a marked degree.

3 The greatest loss in burning fuel oil undoubtedly comes from careless manipulation of draft openings and of the damper. Much of the writer's experience has been in the testing of irrigation plants of from 50 to 600 boiler h.p. capacity where the load was constant and the efficiency in the larger plants ought to have been high. Many of the tests were made to determine the actual conditions of ordinary running without offering any suggestions as to the best way to improve economy. In a great many plants the firemen leave the draft doors wide open and so long as no smoke appears they are satisfied, never questioning whether the amount of air is greatly in excess of that required for economical results. Under these conditions, boiler efficiencies ranging from 55 to 60 per cent are to be expected and the results show that such efficiencies are obtained. On the other hand, when acceptance tests of plants were made under a fuel guarantee and care was used in finding the proper draft area, and especially in the manipulation of the damper to maintain as high a pressure as possible in the furnace and around the heating surface of the boilers, the efficiency has ranged from 73 to 75 per cent.

4 The amount of fuel saved by an intelligent control of the furnaces is therefore a large item in the annual fuel bill. This fact has

not been sufficiently brought home to the operators of power plants using fuel oil. No doubt there is much to be learned regarding the control of boilers using this fuel, and there is still a wide field for research work having for its object the conserving of fuel in the form of crude oil and pointing out a few simple rules to guide the operators of such boiler plants towards a more economical generation of steam.

THE AUTHOR While the interest in this paper as evidenced by the discussion is very gratifying, it is disappointing that for purposes of comparison specific statements of economy have not been given for some of the more important Eastern and Western power stations.

2 During the last six or eight years, the steam-power central station has experienced apparently a wonderful development, and the fuel economy of these large stations is a matter of supreme importance to mechanical engineers. With modern systems for central-station records and cost-accounting, it cannot be that, privately, this subject has not received its full share of attention and study; yet, with perhaps one exception, the engineering public might search in vain the proceedings of our leading engineering societies, and even the technical press, for an authoritative statement of plant fuel economy in regular every-day service in any of these notable central stations.

3 During recent years there have been pronounced differences of engineering opinion regarding the selection of prime movers, and while these differences do not center on the question of fuel economy alone, this is, in most cases, a prime factor. It has become a custom in the selection of such apparatus to compare the steam consumption guarantees for various types at specified ratings, making reference to actual shop and plant tests as a warrant for the guarantee. It is a fact, however, that the final measure of economy is in the monthly maintained station fuel records, under actual service conditions and covering a wide range of load, rather than in shop or uniform-load tests. The engineering public are far more interested in these commercial fuel economies, than in mere statements as to steam consumption under ideal test conditions, or in speculations as to probable fuel economy based on an assumed steam economy; and in passing judgment on a matter of such importance, they are entitled to such fuel performance records.

4 The results of the official fifteen days' test of the Redondo plant were obtained on a commercial run under rapidly swaying railway

load, and including all standby losses for each day during the four and one-half hour period of shutdown. It is common knowledge that there is a large discrepancy between such commercial load records and shop or uniform-load tests, a feature apparently overlooked by some members in discussing this paper.

5 The Redondo station operates in parallel with a waterpower station of about equal capacity. Practically all of the total load-variation is taken up by the Redondo engines, and as these swings correspond to those of a plant of double capacity, it follows that the rate and magnitude of load variations on each unit are about twice as great as on an ordinary or independently operated steam plant.

6 The writer has been keeping careful note of the monthly economy of the Redondo plant, in order to detect any failure, chargeable to the plant equipment, to maintain the economy as shown at the time of completion. There are external conditions—principally the intermittent failure of circulating water due to seaweed, and the operation of all or part of the plant non-condensing for intervals varying from a few hours to a few days per week—which for certain months have caused a falling off of the monthly economy by not more than from five to ten per cent.

7 But to answer properly Mr. Ennis, who apparently is under the mistaken impression that the record given was merely for a test condition, and who considers the economy stated "perhaps unprecedented if it is maintained in the ordinary operation of the plant," the writer can state that for the month of December, 1908, after seven months' operation of the Redondo station entirely under the direction of the regular plant operatives, the average economy, under similar conditions of load, etc., was even better than that obtained under the official test of the complete plant. Provided the plant upkeep is given ordinary attention, there is every reason to believe that the station will continue to maintain practically the same fuel economy; except during periods of unfavorable external conditions of the character outlined.

8 Several members have expressed the idea that compared with Eastern coal-burning plants the excellent fuel economy of the Redondo station is to be attributed wholly to the superior boiler-plant efficiency, when burning crude oil, and not to the economical performance of the prime movers, which, either by inference or conjecture. they would class as ordinary.

9 It is important to distinguish between the high efficiencies resulting from an absolutely uniform-load boiler test under ideal

conditions permitting refined adjustment, and the performance under constantly changing railway load, with heavy swings and including not only a considerable period of operation at light load, but several hours of standby and radiation loss during each 24-hour period; and however good the gross boiler efficiency may be with oil fuel, there results a lower net boiler-plant efficiency, after making deductions for the steam consumption of oil burners, heating coils in oil-suction tanks, oil-pressure pumps and live-steam make-up in oil-pressure heaters, and after making the further uncertain allowances in the case of excessive moisture and sulphur in the oil, and loss by vaporization due to heating in suction tanks.

10 Judging by various general statements, there unquestionably prevails an exaggerated idea of the difference between this net boiler-plant efficiency when burning oil fuel, and when burning the best grades of anthracite and semi-bituminous coals, under the every-day commercial service conditions described, and even when using the automatic system of oil-firing. In short, the economy of this plant in view of the necessary losses in its various elements, under regular service as compared with test-conditions, indicates that not only was the boiler performance remarkably fine, as correctly indicated in the discussion, but that the individual performance of every element composing the plant was necessary of notable excellence.

11 In line with these considerations, Mr. Barrus and others have inquired for test data on the various elements of the plant. The only official tests made at Redondo are those given in the paper, and made under the control of the Testing Committee appointed by the contractor and purchaser. The only tests of individual elements were informal tests of short duration, in which the purchaser was not represented, and although they might be of service in estimating the performance to be expected in designing new plants they would not add to the value of the paper, when limited to the purpose for which it was written.

12 It seems to have been taken for granted by some that the writer is an enemy of the steam turbine, which is not so. All he desired was to bring out the fact that the piston steam engine is still an important factor in steam power plant design. On the present showing of Pacific Coast plants, he is compelled to favor the piston engine on the score of economy, but he does not wish to take a stand that either one or the other, or a combination of the two, will necessarily be the most efficient, when all the elements which affect the every-day working of a power plant are considered. Under constant

full-load "test conditions" the steam consumption of engines and turbines may be nearly equal, but under commercial service and on widely varying loads there may exist a marked difference in the respective fuel-economy records, and fuel economy is really "the proof of the pudding."

13 In answer to Mr. Kent—colder circulating water can be had in the East than in the West, and as the economy of the turbine depends largely on the vacuum obtainable, turbine plants installed in the West with 70-deg. circulating water will not show as favorable economy, either in steam consumption or the "coal pile," as plants in the East, where circulating water is available for a season at 35 deg. fahr., and where a vacuum of between 29 in. and 29½ in. is then obtainable.

14 Mr. Ennis states that the best records for Eastern coal-burning turbine plants are between two and two and one-half pounds of coal per kilowatt hour. If, as is probably the case, these stations are burning the best qualities of anthracite and semi-bituminous coal, then after making due correction for boiler efficiency, the Redondo station will still be ahead on the score of economy.

15 In estimating the probable economy of the Redondo engines, Mr. Ennis used a combined efficiency of engine and generator of 0.855, popularly assumed as a representative figure for steam engines, whereas that actually observed and stated in the paper is over 0.94.

16 Referring to the deduction of Mr. Bibbins, the writer again wishes to protest against theoretical considerations and calculated economies, either from prime movers to complete plants, or *vice versa*. Complete plant economy is not merely a product of engine and boiler performance, but involves many other well-known items of loss. There must be considered not only the uniform load test performance of boilers and engines at economical load, but also their poorer performance in the station on variable load, including light load and overload; and all other elements of station fuel-loss, including standby, starting and stopping, will decrease the actual every-day station economy, as compared either with the calculated full-load economy or that shown by a test of short duration. Hence, knowing the variable load economy of the complete plant, and say, the boiler efficiency, the calculated steam consumption of prime movers will be too great if these other items are neglected. Also if the boiler and engine test performance are known, the calculated complete plant economy is far too good if the other items

of loss are neglected. Comparing with actual service records, Mr. Bibbins' calculation of one kilowatt-hour on fuel-consumption of 21 000 B.t.u. offers an example of the confusion resulting from such false analyses. If Mr. Bibbins and Mr. Kruesi would present to the Society some actual station fuel records, of their respective companies, either for Eastern or Western turbine plants, they would add to our store of such information, and perhaps encourage others to come forward with the same sort of data.

17 The writer's suggestion, commented on by Mr. Ennis, as to the establishment of a standard for comparison of fuel economies for power stations, had reference to the extension to complete plants of the standard set forth in the report of the committee of the British Institution of Civil Engineers, which referred only to the performance of steam engines, under different initial and final pressures and temperatures. An extract from this report is given in Ripper's The Steam Engine, Theory and Practice, p. 300. This cycle is designated by the above committee as the Rankine cycle, and is best known under this name, although previously known as the Clausius cycle.

18 The writer would emphasize the important difference in application between the standard proposed by the committee of the British Institution of Civil Engineers, and that proposed by the writer. The former refers to steam motors only, and the lower limit of temperature is that measured in the exhaust pipe near the engine. The latter refers to complete plants, including steam motor, boilers and auxiliaries, and not merely the prime mover; the lower limit is the initial temperature of circulating water and not the temperature in the exhaust pipe. The complete plant performance will depend on the efficiency of the condensing plant, and the nearness with which the vacuum approaches the lower temperature limit established by the entering circulating water.

19 Referring to Mr. Bibbins' remarks on the performance of engines at fractional loads; owing to the increasing number of expansions, and the benefits derived from the use of superheated steam, the Redondo engines actually improve, over rated economy, for a wide range of fractional load. And contrary to the prevailing opinion a good vacuum is very desirable for high engine economy, the benefit on fractional loads being especially notable. This improvement in engine economy at the lighter loads was so marked that the fractional-load losses in the remaining portions of the plant were equalized, giving for a fairly wide range of load a practically uniform fuel econ-

omy for the complete unit of engines, boilers and auxiliaries. The popular conception of the performance of the average engine on fractional load is so decidedly contrary to this, that the writer feels that he has established a significant fact in giving the curve of complete plant fuel performance.

20 To the writer's knowledge there was no consideration of increased flexibility in the inception of the four-cylinder engine, but this feature is not therefore of any less importance. The combination of the four-cylinder engine with twin condensers combines features of the utmost value, when considering the continuity of station output with a limited number of units, and also when operating in locations necessitating frequent cleaning of or tube renewals in surface condensers.

21 For the frequent special conditions at Redondo, of failure of circulating water and stoppage of condensers by seaweed, the twin-condenser arrangement is of the utmost value, permitting the cleaning of each condenser, while the entire engine exhausts to the other, thus losing between one and two inches of vacuum only. When the supply of cooling water is completely stopped by seaweed, the ability of the engines to carry the same heavy overload non-condensing, and this with high economy, is likewise a feature of great value, for on such service the effective capacity of any prime mover is its capacity when operating non-condensing.

22 Mr. O'Neil is apparently surprised, not that the actual result was so good, but that the guarantee was so poor, and he feels that this difference reflects on the ability of the engineers to determine the guarantee. This guarantee was not based on a calculated economy; it included the elements of standby loss, and variable load running, on which no data were publicly available. On the other hand, the guarantee was based on what information could be secured as to the actual performance of existing stations burning oil fuel. The contractor's initial proposal stated that the calculated economy of the complete station based on individual guarantees of apparatus, and uniform load at rating, would be 240 kw-hr. per bbl. of oil. As the existence of the Redondo station was dependent on the guaranteed cost of power, under commercial conditions of operation, its general manager later demanded a commercial load guarantee subject to a heavy penalty for failure.

23 Answering Mr. Moultrop's query as to the method of filtering cylinder oil from the feed-water, and the possible trouble from oil in the boilers, on account of the use of condensed engine exhaust as

boiler feed-water; owing to the limited supply of feed-water, it was impossible to break in the Redondo plant on raw water, and to throw the air-pump discharge overboard. An excess of cylinder oil was intentionally used in starting the engines, some of which passed by the feed-water filters and for a period caused the burning out of some of the lower row of boiler tubes. The quantity of cylinder oil was subsequently reduced so that the filters could remove enough oil to avoid difficulty with the boilers.

24 Most of the cylinder oil used by the engine is fed into the high-pressure cylinders, and this is largely eliminated by the oil separator diaphragm on entering the engine receiver, and trapped to waste. The steam entering the low-pressure cylinder is consequently comparatively free from oil. As but a small amount of cylinder oil is directly fed into the low-pressure cylinders, the total amount of oil in the condensed water is relatively small and as easily cared for by the filters. The filters are not of the pressure type, but receive water under atmospheric pressure, which passes downward through the filter by gravity. There are two filters per engine, each about 4 ft. in diameter by 5 ft. in height. In each filter are four horizontal perforated diaphragms for the support of the filtering material—usually hay—with which the filter is completely filled. The discharge pipe is so arranged as to submerge constantly all of the filtering material.

25 This general plan of filtration is in use in all Pacific Coast steam engine power plants having surface condensers, and has been successful for a period of years, although the same result does not seem to have been accomplished in Eastern power stations.

26 In view of the scarcity of accurate data as to power-plant performance, and the great difficulty of getting authentic reports tending either to prove or disprove the claims for various types of prime movers, it is to be regretted that a commission of disinterested engineers has not been appointed, to coöperate with some government bureau in obtaining complete data pertaining to the economy of all important power stations. The report of such a commission would officially stamp the correctness, or the incorrectness, of the economies claimed.

27 The writer is conversant with conditions in many Pacific Coast plants, and would be glad to coöperate with such a commission.

No. 1215

EFFICIENCY TESTS OF MILLING MACHINES AND MILLING CUTTERS

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Member of the Society

The design of standard machine tools was for many years, and has been until quite recently, a matter of practical experience, judgment and intuition, and might as well have been called guesswork. Special tools of unusual magnitude were built on somewhat more scientific principles, and this, perhaps, simply because the required dimensions were quite out of the scope of the designer's experience. It was not quite so easy to show judgment about a 6 in. as a 1½ in. lead screw; and it would not have done at all to make the screw 12 in. just to make sure; though the instances are not at all rare when a machine part, screw or shaft, on some smaller machine was made double the required size, simply to be on the safe side.

2 This condition was not due to ignorance on the part of designers or inability to apply engineering data, but to the fact that such data did not exist or were not public property when they did exist. Further, there was no inducement for the machine tool builder to spend time and money collecting data, when the tools, as built, filled all requirements to a reasonable extent.

3 This condition ceased to be satisfactory with the coming of the electric motor into the field of the machine tool builders. At first the wildest and most varied guesses were made as to the size of motor required to drive a certain tool. One builder would supply his machine with a 2½ h.p. motor, while another would equip the same size and style of machine with a 15 h.p. motor. This chaos was made worse by the fact that most purchasers of motor-driven tools would specify the size of motor to drive the machine. One user would condemn the machine tool builder because the 10 h.p. motor

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

was entirely too small and the next one because a 5 h.p. motor was much too large, for the same machine.

4 This brought before the machine tool builder the fact that a machine was not always used to its full capacity, and was sometimes badly overloaded. In sheer self-defense the machine builder was compelled to determine for himself the proper size motor to be put on his machine, and he began to make tests and collect data. Crude as most of these tests were, they laid the foundation for a reasonable method of machine tool design.

5 Before some semblance of order was created out of this chaos, a second disturbing factor made itself felt: high speed steel.

MEASURE OF MACHINE TOOL PERFORMANCE

6 The first attempt of the machine tool builders was to determine the proper size of motor for a given tool. The difficulty they met was to determine when a thing was proper. The substitution of a motor for a belt was not the simple matter it looked. For years past machine tool designers almost uniformly credited a double belt with a pulling power of 50 lb. per inch width. Tests showed that a belt would do twice and even three times as much, though this was bad practice as far as the belt itself was concerned. Another factor in the calculation of the horsepower was the speed of the belt; but what was the speed, when the cone had four or more steps and the countershaft two or more speeds? And even if the belt power could be calculated, was this the proper power for the machine? Would the machine stand all that the belt would stand? Or again, was the driving power all the machine would stand? In other words, was there a proper relation between Power and Rigidity?

7 This showed the necessity of establishing some standard which would be a measure for the performance of the machine, a basis for calculating the size of motor or other source of driving power, and a gage to which the different functional parts of the machine could be proportioned. At first the pressure at the tool point was taken for this standard. A lathe would be specified as able to develop and stand a pressure at the tool point of, say, 6000 lb.; a planer was to have a pull at the table of 20 000 lb., etc.

8 This might have been a good standard if it were possible to determine the pressure on an ordinary machine. Tests were made to find the pressure required for various cuts, but these tests required

a special arrangement of tool post and could not be taken without modifying the construction of the machine. Experiments showed marked differences in horse power required to produce a given pressure at the tool post for a given speed of the work; in other words, the efficiency of a machine of given type and size was by no means a constant. This was no surprise, of course, but had been overlooked in some unaccountable way. In fact, it is being overlooked even now, and the idea seems to be abroad that each type of machine has its constant, expressing the ratio between energy delivered to the machine, and useful work done by it.

9 The difficulty of determining the pressure at the tool post required for a given cut led to the fairly general adoption of another standard, the amount of metal removed per minute. The whole matter is still in an unsettled condition. Specifications for machine tools calling for a guaranteed output are by no means general, but the general tendency seems to be in this direction. It is customary nowadays to call the buyer's attention to the size of driving pulley and width of belt and to the gear ratio, rather than to the output of the machine. This is misleading. Even if actual belt power—that is, the product of belt width and belt speed—were given, instead of size of belt, this would not indicate the capacity of the machine for removing metal; while gear ratio has absolutely no significance except to indicate the reduction in speed which one can obtain in a certain machine. It may be possible that some day in the future, machine tool builders will adopt a set of rules by which to indicate the capacity of a machine, and the first item would doubtless be the amount of metal removed per minute.

10 Though numerous tests have been made as to performance of machine tools, they have been confined almost exclusively to lathes. All other machines seem to have been considered as following the lathe in cutting characteristics.

SPECIFICATIONS OF MILLING MACHINE TESTS

11 When the Cincinnati Milling Machine Company intrusted me with the design of their line of horizontal and vertical high power millers, I found available a great mass of valuable data collected during the existence of this firm, but in order to start with a clean slate and unhampered by past experiences, I disregarded for the time being all these as well as all other data within my reach.

12 The main points to be settled were:

a How much metal shall a machine of given size be capable of removing?

b How much power is required for this work on existing machines?

c Is it possible to improve on the efficiency of present machines and still produce a commercially successful machine?

d How much power is required for the feed?

e What is the efficiency of the feed mechanism?

13 The first question was a point to be decided by the sales manager rather than by the engineer, as it was greatly a matter of competition. As is quite usual in the design of a new line of machines, the desired capacity was placed somewhat higher than that of other makes of similar machines. The customary method is to increase the size of driving pulleys or belt, or the gear ratio, or any combination of these elements. This was not deemed advisable, for two reasons. In the first place, nothing was to prevent competitors from going still further in this matter and so securing a seeming advantage in the sale of their machines without sacrificing anything but a few pounds of cast iron. In the second place, the purchaser of the machine is not interested in the amount of power which he can feed into the machine, but rather in the power of the spindle end, which to him is the business end of the machine. There can be no doubt but that it is better for the user of the machine if he can take a given cut with less belt or motor power. Not only does he save in power, but this power saved is power which would otherwise have been used to wear or destroy the machine.

14 To determine how much power was required to remove a given amount of metal, tests were made on various makes of machines. The metal to be cut was in all cases, both in these tests and in those to be described later, steel of the following specifications:

Combined carbon	0.16 per cent
Silicon	0.008 per cent
Manganese	0.51 per cent
Phosphorus	0.086 per cent
Sulphur	0.041 per cent

Tensile strength per square inch	52378 lb.
Limit of elasticity	30313
Elongation per cent of length	50 per cent
Per cent reduction of area	54 per cent

15 The test blocks used were 18 in. long, $5\frac{1}{2}$ in. wide and $5\frac{1}{2}$ in. thick. The ends were milled in to provide means for holding the block on the table of the milling machine. In all tests a spiral cutter with nicked teeth was used, $3\frac{1}{2}$ in. in diameter, 6 in. face, and for a $1\frac{1}{2}$ in. arbor. The cutter was driven by a key, and made of high speed steel by the Union Twist Drill Company of Athol, Mass. All tests were made by driving the machine by an electric motor, belted to the machine. The object was to have all conditions as near as possible to those under which the majority of milling machines have to run, the only difference being that the belt was nearly horizontal instead of vertical. In testing the efficiency of machines in this way the belt must be considered part of the machine. The power consumed was ascertained by reading of ammeter and voltmeter, and the amount of metal removed by measuring width and depth of cut and the amount of feed per minute. The amount of feed, as indicated on the index plate of a milling machine, is generally an approximation, near enough for every day work but not for a test. The actual amount of feed was therefore computed from the gearing. In case the feed is a function of the spindle speed, that is, if the feed is expressed in thousandths of an inch per revolution, the feed per minute depends on the number of revolutions per minute of the spindle as well as on the amount of feed per revolution. The exact number of revolutions of the spindle at normal motor speed was determined by computing the gear ratio and pulley ratio. As pulley diameter the diameter of the pulley plus thickness of belt was taken. This computation would give the spindle speed at rated motor speed and without belt slippage. For this reason the speed of the first driving shaft of the machine was determined by the tachometer at each test. All readings were taken simultaneously. The number of revolutions of spindle and the feed per minute were corrected according to the tachometer reading.

16 Where the feed of the machine was independent of the spindle speed, that is, taken off a constant speed shaft, and therefore expressed in inches per minute, the same correction was made, as the amount of feed and therefore the amount of metal removed; here again was a function of the speed of the first driving shaft.

17 These preliminary tests showed considerable differences in the efficiency of different makes of machines, that is, one machine would cut considerably more material than another for a given amount of horsepower developed by the motor. They also showed that the efficiency of all machines was relatively low as compared

to the lathe. This latter might have been expected considering the nature of the cutting tool. As the main problem in a machine shop is not to save power, but to get the greatest possible output, this lack of efficiency cannot be held up against the milling machine as a type, for its other peculiarities make it highly efficient as a producer of work. The fact that one make is so much more efficient than another is of great importance, however. It shows that the less efficient machines:

a Use a needlessly large amount of power.

b Have less capacity than they might have for removing metal.

c Use a large amount of power constantly for the purpose of breaking down the machine.

18 It follows that it should be the aim of the designer to produce a machine of the highest possible efficiency as a power transmitter, because

a This insures economical use of power.

b It increases the capacity of the machine.

c It insures long life and freedom from repairs.

19 This high efficiency requirement stands by itself. It must be supplemented, however, by other good features, such as convenience, etc. This paper deals with the efficiency problem only, and that only so far as to show results obtained, rather than the means by which they were obtained. It may be stated, however, that the main features aimed at, and considered essential to high efficiency, were the following:

a Absence of combined torsional and bending stresses in shafts.

b Absence of torsional stresses in shafts subjected to heavy loads.

c Moderate gear speeds.

d Moderate shaft speeds.

e Minimum number of gears in action.

f No gears rotating, which are not required for the transmission.

g Tumbler bearing anchored solidly, and not merely hung from a lever.

h Pulley shaft relieved of belt pull.

There was nothing radical or revolutionary in these points, but it was thought that strict adherence to well known principles should give the best obtainable results.

20 More complete tests were made after the line of high power millers was developed and completed. These tests were of three kinds:

a Tests determining the amount of metal removed per horse power.

b Tests determining the efficiency of the feed mechanism.

c Tests determining the efficiency of the driving mechanism.

21 Three makes of machines were used, the new style Cincinnati High Power Miller, and two other makes, which for obvious reasons cannot be named here, and will be indicated by B and C. The sizes compared were the B No. 4 Plain, C No. 3 Plain, and Cincinnati No. 3 and No. 4 Plain, which will hereafter be indicated by the symbols 3-A and 4-A.

POWER REQUIRED TO REMOVE METAL

22 The same motor and belt were used for all cutting tests. A series of tests was made with a depth of cut of $\frac{1}{16}$ in., $\frac{1}{8}$ in., $\frac{3}{16}$ in., $\frac{1}{4}$ in. and $\frac{3}{8}$ in. This complete test was repeated four times. The cutter was sharpened in the ordinary way before starting a complete series of tests, and not resharpened during the test. The same cutter, resharpened, was then used for the next machine.

23 For each depth of cut a number of different feeds were used. In all cases the even feeds were used starting with the second (next to the lowest) and increasing thus: 2d, 4th, 6th, 8th, 10th, etc., up to the highest feed. It was of course impossible to go through the entire series with the deeper cuts, as the belt would slip before the last feed was reached. This slippage of the belt was at all times the end of the test for that depth, except where the entire scale of feeds could be used. Readings were taken, when the belt slipped, but these readings were not used in plotting curves. They served as a check on the belt tension. It was found that the ammeter readings gradually increased from the first to the fourth series. This was probably due to the gradual dulling of the cutter.

24 In plotting the curves, the test readings were first corrected: the power readings by means of the efficiency chart of the motor; the amount of metal for loss of speed of the machine. The curve as plotted is the curve of the average value of powers. Where the

curve shows an amount of metal removed of say $5\frac{1}{2}$ cu. in. per minute this amount may be due to a depth of cut of $\frac{1}{16}$ in. and a feed of 16 in. or to depth of cut of $\frac{1}{8}$ in. and a feed of 8 in., or perhaps a depth of cut of $\frac{1}{4}$ in. and a feed of 4 in.

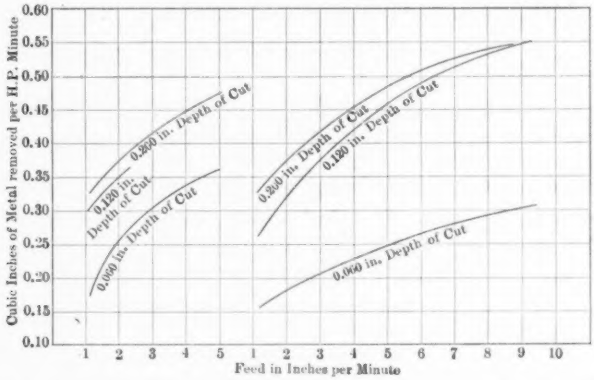

FIG. 1 CUTTING SPEED 12 FT. PER MINUTE FIG. 2 CUTTING SPEED 32 FT. PER MINUTE

WORK OF 1 H.P. MIN. MEASURED IN CUBIC INCHES OF METAL REMOVED, FEED INCREASING

25 The amount of power required to remove a given amount of metal varied with the speed, depth of cut and feed per minute, and seems to have a tendency to the minimum when the section of the

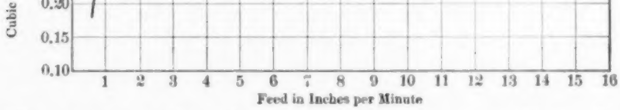

FIG. 3 WORK OF 1 H.P. MIN. MEASURED IN CUBIC INCHES OF METAL REMOVED, CUTTING SPEED 45 FT. PER MINUTE, FEED INCREASING

chip removed per tooth approaches most nearly a perfect square. Fig. 1, 2, 3, 4, 5, and 6 show curves giving relation between power required and metal removed under different conditions of speed,

feed and depth of cut. They are partly derived from the tests described, and partly from tests made for the special purpose of ascertaining these relations. The fact that the power required changes

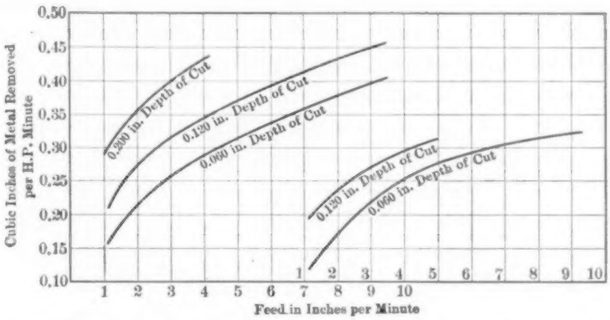

FIG. 4 CUTTING SPEED 67 FT. PER MINUTE
FIG. 5 CUTTING SPEED 150 FT. PER MINUTE

WORK OF 1 H.P. MIN. MEASURED IN CUBIC INCHES OF METAL REMOVED, FEED INCREASING

with these conditions of feed, speed and depth of cut made it impossible to plot a single curve giving the relation between the power and the metal removed. Besides, the same speeds and feeds obtainable on

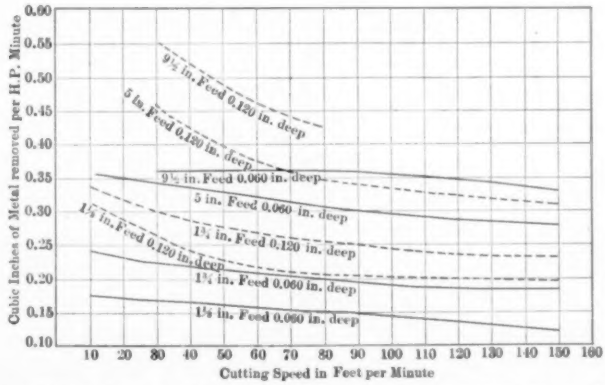

FIG. 6 WORK OF 1 H.P. MINUTE MEASURED IN CUBIC INCHES OF METAL REMOVED, CUTTING SPEED INCREASING

one machine were not obtainable on another. And again, the amount of slippage or slowing down of the machine was not always the same for the same amount of metal removed. For this reason the averages have

been plotted, and these averages were obtained for the various machines in substantially the same manner. The curves were extended to the zero point, but the high point of all curves is the actual highest average obtained, so that in a certain sense the curves also show the comparison of the greatest possible capacity of these machines. This should be taken as significant, however, only when remembering the conditions under which the machines were tested, and then only as a measure for the maximum driving power.

26 No part of the A machines showed undue stress during the test. The B machine was strained in feed and drive parts, but showed sufficient rigidity in its other parts, while the C machine

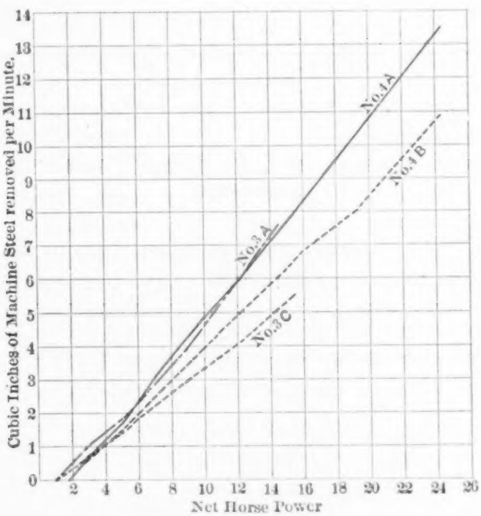

FIG. 7 CUTTING EFFICIENCY CURVES

was apparently in distress. It is only fair to say that all three machines were worked up to a limit far beyond what they would be called upon to do in daily operation. It goes to show, however, how utterly foolish it would be to furnish these machines with larger pulleys or wider belts for the purpose of giving them more capacity, as they can do much more work, even with their present pulleys and belts, than some of the other parts will permit. Multiplying the product of belt width and speed and calling this 100 for the 4-A machine, then the corresponding product for B No. 4 is 144. Calling this same product 100 again for No. 3-A, the corresponding product

for the C No. 3 machine is 154. This shows clearly again the small value to the buyer of the size of driving pulley or belt power, and once more shows the necessity for machine tool builders to come to some mutual understanding as to what should be their selling guarantee in regard to power.

27 Fig. 7 shows the curves obtained, the abscissae being the number of horse powers net and the ordinates the number of cubic inches of metal removed per minute.

POWER REQUIRED FOR FEED MECHANISM

28 Tests made by the Cincinnati Milling Machine Company, as well as by other concerns engaged in the manufacture of milling machines, had shown that a considerable amount of power is required for the feed drive of a column and knee type of machine. It was found that as much as 40 per cent of the total power applied might have to be used for the feed alone: and that this could amount to as much as $3\frac{1}{2}$ h.p. on a No. 4 machine. Inasmuch as I did not conduct any of these tests and do not know all of the conditions under which they were made, I do not feel that I can present the results. The main result, however, as far as I was concerned, was the fact that they made me give very careful consideration to the feed mechanism. If $3\frac{1}{2}$ h.p. is used for the feed, and if that feed is 20 in. per minute (the highest feed found on a modern milling machine), then the pressure against the cutter must be

$$\frac{3\frac{1}{2} \times 33000 \times 12}{20} = 69\ 300 \text{ lb.}$$

if there are no losses in transmission. Further, it is not likely that the greatest amount of horse power is required for the feed at its maximum number of inches per minute. It is more probable that the maximum of feed power is used for 10 in. feed or less per minute, in which case the pressure would be in the neighborhood of 140 000 lb.

29 A machine using 6 h.p. net for the drive alone (that is, using 6 h.p. at the cutter) would require about 8 h.p. at the pulley for the drive alone, and therefore a total of $11\frac{1}{2}$ h.p. for drive and feed under conditions of feed mentioned above. This is quite a respectable amount for a No. 4 machine. Assuming the cutting speed to be 40 ft., the pressure at the circumference of the cutter must be

$$\frac{6 \times 33\ 000}{40} = 4950 \text{ lb.}$$

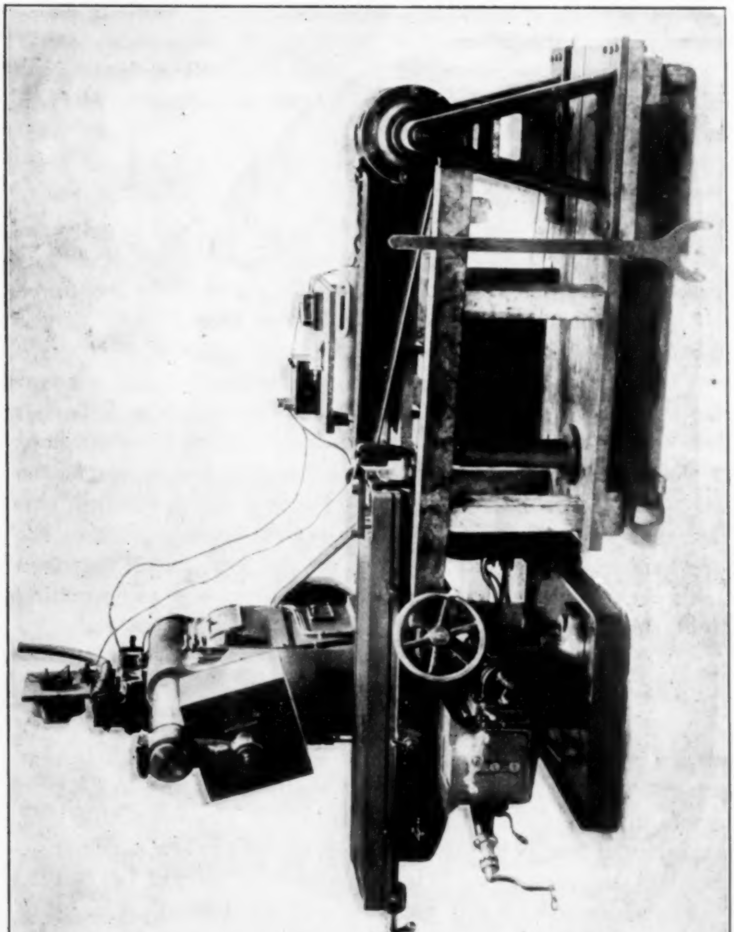

Fig. 8 Apparatus for Testing Feed Efficiency of Milling Machine

and this must also be the approximate pressure against the table screw instead of 140 000 lb. Of course all these figures are assumed, but they illustrate the computations which led up to the second line of experiments to be described here.

30 In order to determine the efficiency of the feed mechanism, the amount of power used was measured, as well as the amount of work done by the table. As the amount of power required varied widely, and it would be impossible therefore to obtain an efficiency chart for some small motor, covering the entire range of powers used, the idea of using an individual motor for the feed alone was abandoned. Instead the same motor was used as had been used for other tests. The efficiency of this motor was not known below ¼ load. It was then necessary to provide an artificial and constant load for the motor. For this purpose the square box shown in Fig. 8 was mounted on the spindle arbor. It had a paddle inside and a number of obstructions which made the required resistance for the water in the box. By increasing the amount of water in the box, and by giving the paddle various speeds, any load could be produced within the range of the apparatus, and this load was constant. The box was kept from rotating by resting against the overarm of the machine. The dead load was adjusted until it came within the efficiency curve of the motor. The part of the apparatus used for measuring the work done by the table consisted of a dynamometer, graduated and calibrated up to 8000 lb.

31 One end of this dynamometer was attached to the table of the milling machine. A chain at the other end of the instrument was wrapped around a drum which was mounted on a brake of the Weston type. An arm, mounted on the brake casing and visible in the drawing had at one time been used to determine the efficiency of the driving mechanism but was not used in this experiment, except to form an abutment for the brake. The brake itself was used only as a safety device, being set at sufficient pressure for the test, but not enough to wreck the machine. It had been intended originally to set the brake for a certain pressure and to have the table feed under this pressure for some length of time, but it was found that the brake was too jerky in its action. The mode of working was therefore changed and the test was carried out as follows: All even feeds were taken, beginning with the second lowest and going up to the highest. For each feed the following pressures were selected: 1000, 2000, 3000, 4500 and 6000 lb. The entire series was repeated once if nothing happened to spoil the test. If something happened

to disturb the proceedings, or if something had to be readjusted, then the entire result of the test was dropped and no curve was plotted until two complete series of tests were made without interruption. A sensitive ammeter reading $\frac{1}{10}$ amperes was used. The tachometer served again to take the speed of the first driving shaft, and all corrections were made as in the test previously described. With a given feed, an entire series of pressures was gone through, after which the amount of feed was changed for the new series, etc.

32 Before taking a series of readings, the feed mechanism was disconnected to get the reading of the dead load. Another reading

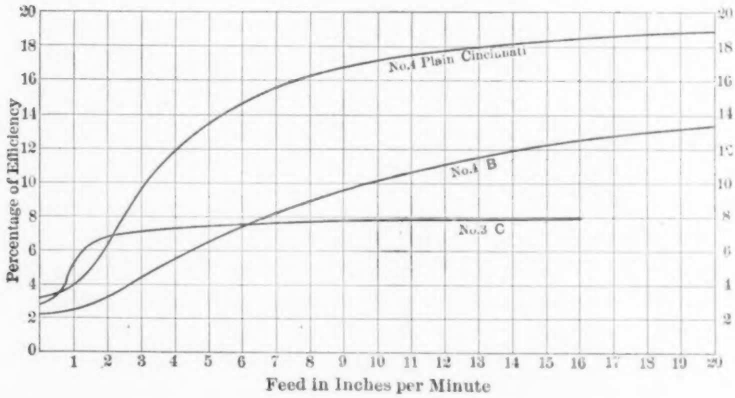

FIG. 9 FEED EFFICIENCY CURVES

of the dead load was taken after each series of readings was completed: that is, after all the different pressures had been used for one single feed. One observer took the readings of the tachometer; another of the ammeter; a third of the dynamometer; and a fourth was stationed at the feed starting lever. The observer of the dynamometer watched his instrument until the pointer reached the proper figure, when he would call out sharply. At this moment all readings were taken and the feed disconnected. Several preliminary tests were made to see if the readings could be duplicated in this fashion, and it was found to be very easy to do so. The rise in the ammeter readings was gradual, and allowed the observer to take the correct reading every time within a tenth of an ampere. If there

was any drop in the speed of the machine, this drop was also very gradual, permitting the tachometer readings to be easily duplicated. The only point not so easily duplicated was the reading of the dynamometer, especially at the faster feeds. It often happened that the feed pressure would go somewhat beyond the pressure intended, but the dynamometer did not change its position when the feed was thrown out; so the observer could take his reading at leisure, the only difference being that instead of having to plot for say 2000 lb. the curve might have to take care of 2100 lb.

33 Fig. 9 shows the curves of average values plotted from these readings, in which each ordinate is the average of the ordinates corresponding to a certain amount of feed. The results of these tests justify again the precautions taken to obtain an efficient feed mechanism in the new line of milling machines. It may be mentioned here that these precautions consisted in avoiding idle running gears, high gear velocities, combined torsional and bending stresses in shafts, and ill supported and floating bearings, and above all, in the use of quick pitch screws.

34 The importance of a higher efficiency in the feed gearing cannot be over-estimated. It may seem that 20 per cent efficiency is still so low that it makes little difference whether it is this amount or something else. It may seem at a first glance that it is of little importance to the user whether 80 per cent or 90 per cent of the feed power is lost in transmission. But a more careful look at this problem shows the importance of high efficiency very clearly. If 3 h.p. out of 10 are used for feed on a machine of which the feed efficiency is 10 per cent, then 0.3 h.p. is actually used for this feed. If 10 h.p. are used for the entire machine—that is, for feed and drive—then 7 h.p. are left for the spindle drive alone, and therefore the amount used for feed is $\frac{3}{7}$ of the amount used for the spindle drive. With a machine having a feed efficiency of 20 per cent, or twice as much, the amount of power used for the feed will be $\frac{3}{14}$ of the amount of power used for the spindle drive alone, or $\frac{3}{17}$ of the total amount used for the machine. If this amount is 10 h.p. again, then the amount used for the feed will be 1.76 h.p. as against 3 h.p. in the first machine; and the amount left for spindle drive will be 8.24 as against 7 h.p. in the first machine, giving an increased spindle power of 18 per cent available for cutting. Of the 3 h.p. used by the first machine for the feed alone, 0.3 h.p. is usefully employed, while the remaining 2.7 h.p. are employed to wear down the feed mechanism and destroy the machine. Of the 1.76 h.p. used for feed in the second machine,

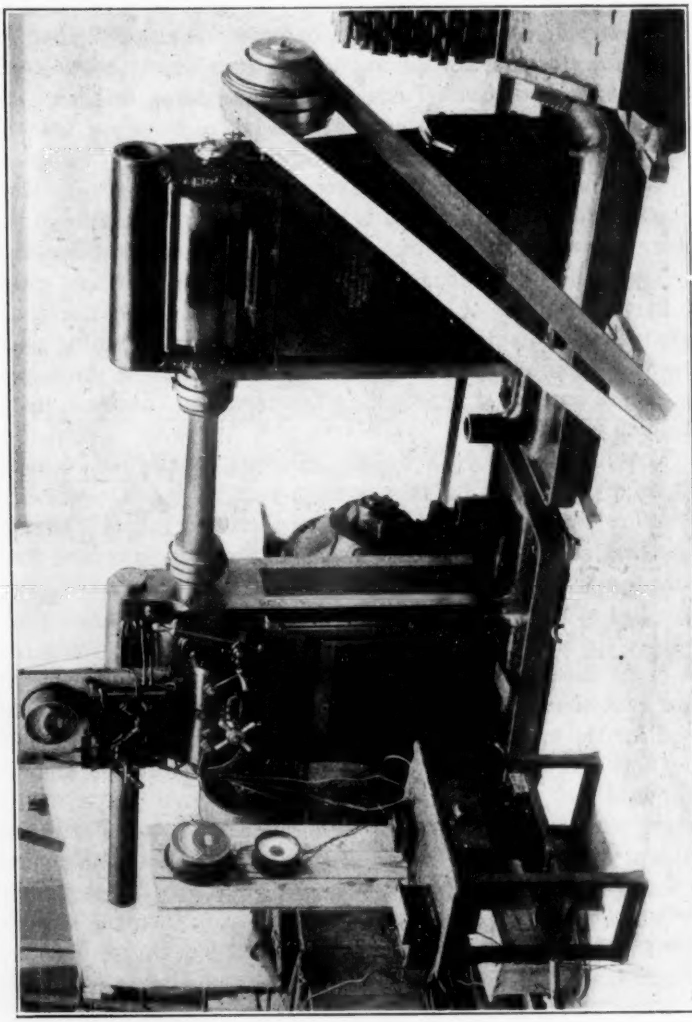

FIG. 10 APPARATUS FOR TESTING EFFICIENCY OF MILLING MACHINE DRIVES

20 per cent or 0.352 h.p. is usefully employed, whereas the remainder or 1.41 h.p. is used destructively. It will be seen therefore that the power used to break down the machine is almost twice as great in the first machine as it is in the second.

TESTS UPON EFFICIENCY OF DRIVING MECHANISM

35 The third series of tests relates to the efficiency of milling machines as power transmitting devices: that is, the ratio of input and output of power. Some preliminary tests were made by observing the power at the spindle by means of an absorption brake of the Weston type; in fact, the same brake as illustrated in Fig. 8. This gave fairly good results at the higher speeds, the torque being small; but at the lower speeds and with greater torque the action of the brake became jerky and it was practically impossible to obtain reliable readings. For this reason the tests were carried on with the apparatus shown in Fig. 10.

36 This consisted of two machines of the same type, make and size—namely, Cincinnati No. 4 High Power Miller—placed opposite each other and connected by a stout shaft. The feed works were removed, as were also knee, saddle and table, so that nothing was left but the bare frames and driving works. The machines were placed with the spindles approximately in line. A flange was screwed to the nose of each spindle; each flange was provided with a tongue engaging the groove in the similar flange opposing it, and keyed to the stout connecting shaft. There was plenty of clearance between tongue and groove and also endwise so that the connection could behave as a universal joint shaft in case the spindles were not exactly in line. It must be remarked here that the motion in this universal joint shaft was exceedingly small. Flat pieces of steel bolted to the first mentioned flange prevented the connection from coming apart. One of the machines was driven by a motor while the other drove a generator. The current thus generated was dissipated in a water rheostat, by means of which the amount of current could be closely regulated. There was a set of electric instruments for motor and generator each, so that all readings could be taken simultaneously. Both machines were driven by belts. A tachometer was used to determine if one of the belts slipped excessively. If the generator voltmeter showed considerable drop and the tachometer showed about the proper speed at the first driving shaft of the first machine (motor machine) then the belt to the generator must have slipped.

If however the tachometer showed a drop, then the belt from the motor to the first machine must have slipped.

37 The speed controlling levers of both machines were always in corresponding positions—that is, both were set for the same speed, so that at whatever speed the spindle was running, both driving pulleys were always running at the same speed, and that is the speed at which they are supposed to run under working conditions. Different sets of readings were taken. In one set the current consumed was kept at as near 125 amperes as possible. In another set this amount was 100 amperes; in still another, 80, and in the fourth, 70. Each set of tests was carried out over a number of speeds, namely, the lowest, third, fifth, etc., up to the highest but one. After correcting the motor reading for motor efficiency and generator reading for gen-

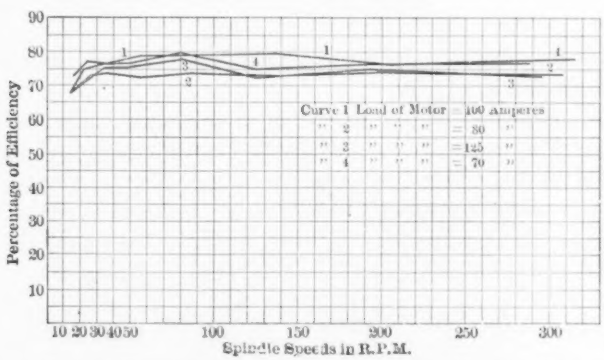

FIG. 11 EFFICIENCY CURVES OF NO. 4 CINCINNATI HIGH POWER MILLER

erator efficiency, the quotient of the corrected readings gives the product of the efficiencies of the two machines.

38 It would be proper to consider the two machines as of equal efficiency if the conditions of load were the same, for these tests did not attempt to establish the efficiency of an individual machine, but of a type of machines, and this could best have been done by submitting a great number of these machines to individual efficiency tests and taking the average of the results. It is therefore plain that a better result will be obtained by considering at once the two machines as having the same efficiency than by attempting to determine their individual characteristics. It was said that it would be proper to do so if the conditions of load were the same for both machines, but this did not seem to be the case. The load was the same only at

the spindle noses. From there on and up to the driving pulley the load increased in the first and diminished in the second machine. The efficiency of the first machine is made up of the efficiencies of its bearings, shafts, sets of mating gears, etc. These efficiencies may be called $e1$, $e2$, $e3$, etc. The total efficiency of the first machine being E, then $E = e1 \times e2 \times e3$, etc. Similarly, taking F as the symbol of the efficiency of the second machine, then $F = f1 \times f2 \times f3$, etc. The f's may be assumed to differ from the e's on account of the difference in load.

39 It was found however on taking the readings that there was such a small difference in efficiency, whatever the load, that it seems to be allowable to consider the efficiency, as practically constant: in which case the efficiency of each machine equals the square root out of output divided by input. The curves presented in Fig. 11 are based on this assumption. It will be seen that the efficiency of the machine varies from 67 up to 79.7. Various tests made at other times have shown repeatedly efficiencies above those plotted in the curves, but it was not thought advisable to embody them in these curves as there were not enough readings to make a complete set and the use of these isolated readings would have a tendency to disturb the effect of the systematic tests.

EFFICIENCY OF CUTTERS OF DIFFERENT TYPES

40 The milling machine is not essentially less efficient as a power transmitter than any other machine tool, but the amount of metal removed per horse power per minute is low; much lower in fact than for the lathe or planer. Were it not for other properties the milling machine could not compete with either of these two machines. It is fortunate for the milling machine that the question of power consumption is of minor importance in considering the purchase of machine tools. Still it cannot be denied that the milling machine would be esteemed higher if its power consumption could be brought down to the level of the lathe. It is obvious that this cannot be done by increasing the efficiency of the mechanism as the margin is not large enough to allow of any material improvement. Any substantial increase of efficiency must therefore be found in improvements of the cutting tool. With this consideration in mind, some tests were made as to the power required to remove a given amount of metal with different styles of cutters. These tests have not been carried out over a long enough period or with a large enough

variety of cutters to give absolutely reliable results, yet, incomplete as they were, they bring out in a startling way the amount of improvement that may be made in this direction.

41 Cuts made with a spiral cutter with nicked teeth showed a best efficiency of 0.48 cu. in. of metal per net horse power per minute, and this efficiency was obtained only in a few isolated cuts and with a very sharp cutter. Cuts made with a 14 in. face cutter with inserted teeth and on a No. 4 Cincinnati High Power Miller showed a production of 0.64 cu. in. per net horse power per minute, an increase of $33\frac{1}{3}$ per cent over the spiral cutter. This result confirms the gen-

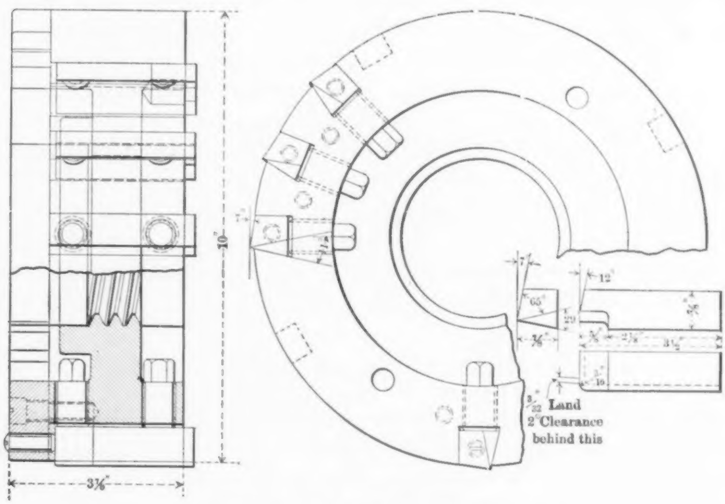

Fig. 12 10-In. 16-Blade Cutter

eral belief that a face cutter cuts more freely than a spiral cutter. The teeth of this face cutter were radial, as it is customary to make them. Tests made with the cutter shown in Fig. 12 showed an efficiency of 0.96 cu. in. of metal per net horse power per minute, and a few isolated cuts even higher. This is an improvement of 100 per cent over the spiral cutter and 50 per cent over the face cutter with radial teeth. The cutter shown in Fig. 13 showed the same efficiency. Both cutters have the blade set tangent to a circle concentric with the cutter, thus giving them a rake angle of 15 deg. The clearance was 7 deg. The points of the cutting blades were rounded to prevent injury by burning or chipping, and this reduced the effective rake

near the horizontal tangent to this curve. It was for this reason that the blades were set leaning backward as in Fig. 13. But for this curvature at the point of the blades, the simpler construction of Fig. 12 would be perfectly satisfactory. The leaning back of the blades is rather a refinement than a necessary improvement.

42 It should be borne in mind that the tests made with these cutters were made on a vertical machine which had the same driving parts as a horizontal machine of the same size, and besides an additional two shafts and four gears, so that it is safe to say that its mechanical efficiency must be less than that of the No. 4 horizontal machine used with the spiral cutter with nicked teeth. The action

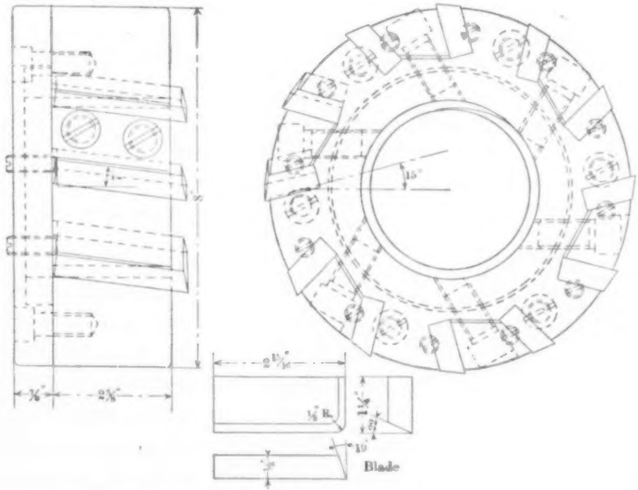

FIG. 13 8-IN. INSERTED TOOTH CUTTER

of the feed on a machine using this cutter may be somewhat different from that on a machine using spiral cutters. Whether there is such a difference and what it amounts to has not been ascertained by tests, and might be a promising field to explore. The fact that, generally speaking, the face cutter with rake removes double the amount of metal with the same amount of power, as compared to the spiral mill, is significant and places the vertical machine in the front rank for slabbing work wherever it is possible to use this type. Equally significant is the fact that a cutter with rake removes 50 per cent more metal than a cutter without rake. It may be mentioned here that many of these cutters, especially for so-called rotary planers,

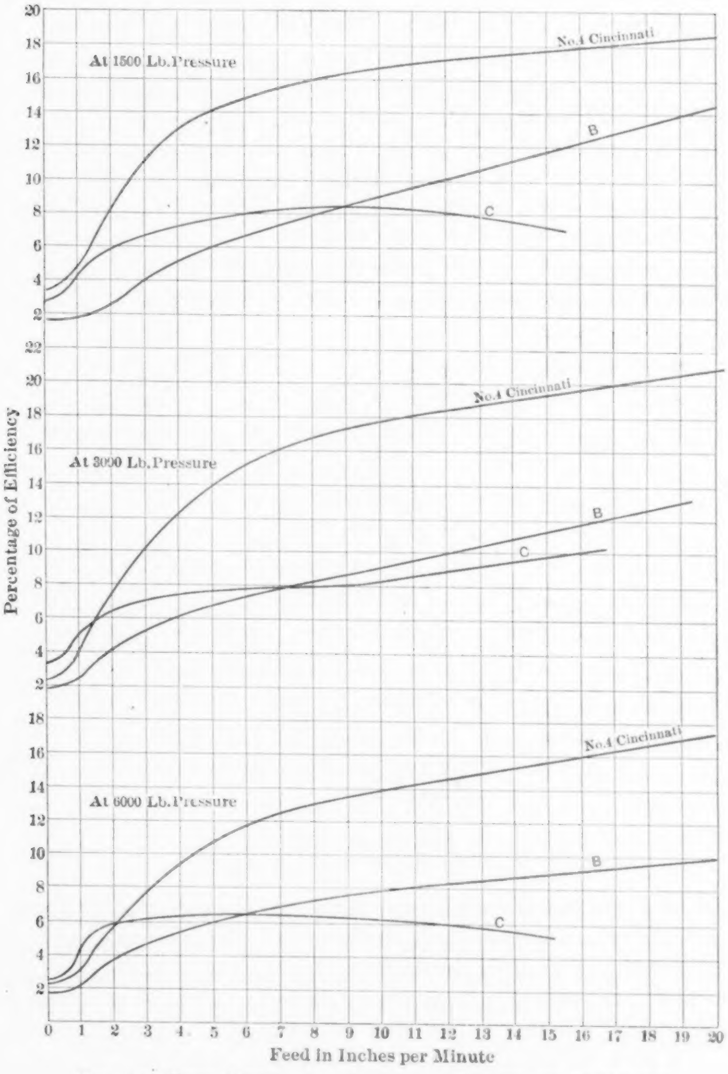

FIG. 14 FEED EFFICIENCY CURVES AT DIFFERENT PRESSURES

are being made and have been made for a great many years with the teeth leaning backward as in Fig. 13 but without rake; the makers and users apparently believing that this leaning constitutes rake.

43 I regret that the requirements of patent laws do not permit me at the present date to describe a spiral cutter and a face cutter built on different principles, which have been tested as to their efficiency. It may be of some interest however to know that a great number of cuts have been taken with these cutters showing removal of 1.14 cu. in. per net horse power per minute, and with entire absence of chattering of machine and spring of the cutter arbor.

44 The writer is fully aware of the fact that the tests described above are by no means complete, and he believes that they are only a starting point and that much more should be done in this line before we can say that we have sufficient working knowledge of the efficiency of drives and feeds of milling machines and of milling cutters. The fact that so little has been published on this subject has led him to submit this paper, incomplete as it may be.

DISCUSSION

Mr. Fred J. Miller Referring to Par. 4, it is, of course, familiar to the author that although the motor drive has emphasized the fact that machine tools are used for widely varying kinds of work, yet long before the electric motor came into use machine tool builders expected their machines to be called upon to do work varying from the roughest and heaviest to the lightest and most refined. Standard, or commercial, machine tools always have been and probably always will be a compromise based upon that fact.

2 A 16-in. lathe, for instance, can be built capable of taking heavy cuts from steel forgings, say 12 in. to 16 in. diameter; but such a lathe would be nearly useless for a great deal of the work that 16-in. lathes are called upon to do. Much the same considerations apply to milling machines and this brings up the question as to whether or not a milling machine of the column-and-knee type should be so built as to take very heavy cuts, or be expected to do so.

3 Machines of this type are more convenient of access than any other and are preëminent for the facility with which they can be manipulated. This fact peculiarly fits them for work of a certain kind,—jobbing and tool work, continually changing in character, for which the machine must be continually changed in its adjustments and

the operator must be able to see easily and clearly what is going on. On such work these features are generally more important than capacity for heavy cuts.

4 Generally speaking, heavy cuts are taken on regular manufacturing work where a number of pieces alike are handled at one time and at a single setting of the machine. On such work facility of access and adjustment, though still important, are unimportant compared with the ability to take a heavy cut when called for.

5 That the knee-type miller is, in its original form, ill-adapted for heavy cuts is shown by the now general adoption of "harness" designed to connect the outer end of the cutter arbor with the outer extremity of the knee and thus reduce the springing and vibration due to reaching out from the supporting column with the cutter arbor and then reaching out with the work-support to meet the cutter. I have seen millers of the column-and-knee type taking surprisingly heavy cuts; but it has never seemed to me that such a machine should be called upon to do it. Where heavy cuts are to be taken machines of other forms seem to be desirable.

6 In Par. 9 of the paper, reference is made to belt power and gear ratio. I have very seldom seen belt speed and gear ratio both specified in a machine tool; it is sometimes done, but I think, not often. Of course if we know the belt width and speed and something of the construction of the machine we can deduce, at least approximately, the turning force that will be applied to the cutter arbor. If we know the belt width only, and the gear ratio, we can do the same thing.

7 Referring to Par. 19, I think it would be desirable to include with the paper a drawing of the driving gear of the machine designed by the author. I believe such a drawing has been published elsewhere and that the construction is not regarded as a secret. It would help very much toward making complete the record here presented if such a drawing were to be included in the paper.

8 Referring to Par. 28 and 29, it would be interesting to know something of how the tests there referred to were made. From the nature of the case we would naturally conclude that the force exerted by the cutter against the feed motion of the platen must be about equal to the force exerted by the feed mechanism to move the platen. In fact, in the case of a very deep cut, a considerable proportion of the force exerted in rotating the cutter arbor is not exerted in directly resisting the feed motion, but in lifting the platen of the machine from its seat. It is a familiar fact that, where a stem cutter or a face cutter is applied to the work in such a way that as much of the cut is above

the center line of the cutter as below it, practically no force is required for the feed, so long as the cutter is sharp.

9 In general we may say that, other things being equal, the force required to feed the platen of a milling machine will decrease per unit of metal removed per turn, or per minute, as the cut is increased in depth. At a depth of cut equal to the diameter of the cutter, it is probable that very little force is ever required for the feed motion, except to overcome friction, unless the cutter be dull. But for wide cuts of no great depth, the conclusion would seem to be inevitable that the force applied to rotate the cutter and the force applied to move the platen against the cutter are practically equal. If these two opposing forces are substantially equal for shallow cuts, and if, as the cut grows deeper, the force required to rotate the cutter increases in a faster ratio than that required to move the platen, as undoubtedly is the case, and if, as mentioned in the paragraphs referred to, the peripheral speed of the cutter is 40 ft. per min., and of the platen 10 in. per min., we have a speed ratio of 48 to 1, and it is obvious that the power consumed ought to be in the same ratio; or, in other words, the power required for the feed should be slightly over 2 per cent of that applied to the rotation of the cutter, instead of over 66 per cent.

10 It is true enough that results of tests are generally to be preferred to those of deductive reasoning, but it would be interesting to know if the tests that showed that 40 per cent of the power applied to the milling machine was used to drive the feed-motion, represent actual practice.

Mr. Wilfred Lewis and Mr. Wm. H. Taylor This paper presents interesting and instructive data in regard to the performance of a milling cutter, and Mr. DeLeeuw is to be congratulated upon the admirable manner in which his work has been done. We have also made some tests upon a milling cutter of different construction, and although we have not gone into the subject in the same way, we are quite willing to accept the results derived by Mr. DeLeeuw in regard to machine efficiencies as fairly applicable to other machines, upon which we have tried our cutter. We are even willing to admit that the Cincinnati milling machine may be more efficient, and think that 60 per cent would come nearer to the efficiency in our case when the motor is included on account of the additional gearing required for the heavier drive. His method of determining the efficiency of his milling machines by coupling two together and measuring the current absorbed and given off is very ingenious, and gives results in all

probability pretty near the truth. We believe, however, that the counter efficiency is never quite equal to the direct efficiency. This is obviously the case where worm gearing is employed, and it must be true to a lesser extent in all cases where speed is reduced through a train of spur gearing and then increased again through a similar train.

2 If Mr. DeLeeuw had carried his experiments further he probably would have determined a higher efficiency for the direct drive of his milling machines. We are willing, however, to accept 75 per cent as a fair average and on this basis it appears that the best results obtained in slab-milling ran from 0.45 to 0.55 cu. in. per min. for one horsepower actually consumed by the milling cutter in cutting 16 carbon steel.

3 In face-milling much better results are obtained and the difference is ascribed to the lip angle of the cutter used in face-milling. It would appear, therefore, that there is no inherent advantage in face-milling over slab-milling if the cutting edges are alike in each case, and our experiments on slab milling bear out this opinion.

4 In our paper on The Development of a High Speed Milling Cutter with Inserted Blades for High-powered Milling Machines, presented at this meeting, we would call attention to the very pronounced lip angle obtained by the use of our curved blades, and as far as the actual performance is concerned we have made experiments since that paper was written, through the courtesy of the Niles-Bement-Pond Company, demonstrating that it is possible to obtain with such a cutter from 1 to 1¼ cu. in. of 25 carbon steel as against ½ cu. in. of 16 carbon steel obtained by Mr. DeLeeuw. This output is estimated on the basis of 70 per cent for the combined efficiency of motor and slabbing machine and this we believe to be a higher figure than could be established by experiments upon efficiency for such a heavily-geared machine. Probably 60 per cent would be nearer the truth and on this basis the amount of metal removed per horse power would run from 1¼ to 1½ cu. in.

5 Mr. DeLeeuw's experiments were made with a 3½ in. by 6 in. cutter on a machine capable of transmitting 11 h.p., weighing about 6000 or 7000 lb. Our experiments were made with an 8 in. by 18 in. cutter on a machine capable of transmitting 165 h.p. and weighing 70 000 lb. We had, therefore, the advantage of taking heavy chips, as well as the advantage of the lip angle referred to, and the limiting capacity of our cutter is not yet in sight.

6 It will be of interest to know that some of these chips (exhibited at the meeting) were removed under more severe conditions than had

before been attempted in slab milling practice with any other type of milling cutter or milling machine. Mr. DeLeeuw states in his paper that the cutter was sharpened before each test. We traversed a steel forging $11\frac{1}{2}$ in. wide and 50 in. long five times without sharpening our cutter and after doing this there was no perceptible dulling of the cutting edges. The first two runs were made at a table advance of $9\frac{1}{2}$ in. and $\frac{21}{32}$ in. depth of cut, and the circumference of the cutter was under approximately 25 000 lb. cutting pressure; the third run was made with a table advance of $5\frac{1}{2}$ in., depth of cut $1\frac{1}{32}$ in., with the cutter under approximately 26 000 lb. pressure, and runs four and five were made with a table advance of $9\frac{1}{2}$ in. per min., depth of cut $\frac{3}{4}$ in. with the circumference under approximately 39 000 lb. pressure, making a total of $30\frac{1}{2}$ min. actual cutting time, in which the cutter traversed 250 in. and removed 607 lb. of steel. The cutting speed in all these tests was 70 ft. per min.

7 In regard to the efficiency of the feed mechanism of the Cincinnati milling machine we are not at all surprised to find it as low as Mr. DeLeeuw has discovered, for the simple reason, that the feed is transmitted through a screw. The efficiency of screws is a matter that has come before the Society and has had very careful consideration. It is well known that a screw is one of the most inefficient mediums for the transmission of power that can possibly be employed. It is nevertheless in many cases the best, but we do not agree altogether with Mr. DeLeeuw that the efficiency of the feed mechanism is a matter of first importance. We believe that the rigidity of the feed mechanism is a matter of more importance than its efficiency. The efficiency can readily be increased by increasing the pitch of the screw. Roughly speaking, the efficiency of a feed screw is measured by the pitch divided by the pitch plus the diameter, and for a screw of $\frac{1}{4}$ in. pitch $1\frac{1}{8}$ in. diameter, such as is used on the Cincinnati No. 3 milling machine, the efficiency should be about 18 per cent. If the pitch were doubled the efficiency would be increased to 31 per cent, and while better efficiency could be obtained by further increase in pitch, this gain in efficiency, however, would be obtained by the sacrifice of rigidity, inasmuch as the steeper the thread, the greater the torsional strain in the screw and the greater the liability to chatter under heavy work.

MR. FRED. W. TAYLOR Mr. DeLeeuw has done admirable work in the experiments which he has described, and work of the kind that is much needed; it is therefore with much hesitancy that I criticise

his paper at all, and my criticism is intended to supplement his paper, not to detract from it.

2 His paper gives us facts as to the relative efficiency of three different milling-machines. He has not given us any data, however, which enables us, as engineers, to judge why one of the machines is so much more efficient than the other two. Now in an advertisement, or even in an article written in a technical journal, it is perfectly proper to call attention to the fact that one machine is far better and more efficient than any of the machines competing with it, without giving in detail the reasons. A paper presented to an engineering society, however, should be for the education of its members, and they obtain but little valuable information or education from the mere statement that a result has been obtained, without indicating the exact means by which the result is reached. Engineers want to know not only the result or effect, but also the cause which has produced it.

3 Now the efficiency of a milling-machine is largely the efficiency of its train of driving-gears with its shafts and bearings, plus the efficiency of the cutting-tool. A full description of the trains of gearing, etc., of the three machines should have been given, so that the readers could decide for themselves exactly why one machine is better than the other two.

4 Again, at the end of the paper is a statement which has the appearance of an advertisement. The author states that a new milling-cutter is being patented which is far more efficient than those on the market, without giving even a view of the cutter itself or the slightest inkling as to the cause of this superiority: such a statement should never be permitted in a paper published by an engineering society.

PROF. J. J. FLATHER I wish to add a word of appreciation of the scientific work which Mr. DeLeeuw has done in determining the power required to remove metal by milling and in subdividing that power, placing a portion of it where it belongs, in the feed mechanism.

2 But I wish to protest against the engineer classing "a matter of practical experience, judgment and intuition," as "guess work," as in the first paragraph of the paper. I think that is an engineering estimate and an inference, rather than guess work, and believe this view is really what the author intended. The design of machine tools has been largely based on such estimates in times past, for lack of accurate data, and the engineer has, by a certain process of intui-

tion, and by using his judgment and past experience, put all these factors together, so as to make a very accurate estimate as to what results would be produced under given conditions.

3 The main object of a machine designer should be to produce a tool that will give increased output. The matter of power is of very little importance. In fact, the cost of fuel in producing power in most of our factories amounts to only two or three per cent, and it does not make a great deal of difference what the power is. What the aim should be is increased output, and any machine that will give increased output, no matter how inefficient it is in the conversion of power, is to be preferred to one that is much more efficient in this regard but has a smaller output.

4 Considering the tools that are in most common use, the lathe and planer and the milling machine, we know, by referring to various tests based on the pound per minute output of metal, that the lathe will remove approximately a pound of steel chips per minute at a cost of 0.4 h.p.; the planer of about 2.5 h.p. per pound; and the milling machine of approximately 10 h.p. per pound under average conditions. There are so many variables that it is practically impossible to predict how much power will be taken, but the tests show the efficiency of the milling machine to be very low; and yet every manufacturer realizes the great saving in labor and time by the use of the milling machine, and the milling machine will be used more and more, because of this saving. This attention to increased output should not in any way prevent us from determining the amount of power required to operate a machine, or its various parts. In fact such tests are often absolutely necessary to a proper handling of the problem; if they lead to a reduction of the power required without interfering with the output, an additional advantage is obtained.

5 The author speaks of the lack of engineering data from which to determine the size of motor to apply to a machine. Today, with the use of high speed steels, this is true to a certain extent, and yet not only have most careful tests been made, to determine the total power required for removing metal and running the machine, but the subdivided power has been determined as long ago as when Hartig was carrying out his experiments in Germany. The experiments showed the amount of power required to run the machine idle, the amount for different speeds, and the amount to remove metal per unit of time. Hartig's experiments included some sixty-nine different machines, with from five to fifty tests on each machine, showing the very wide range of his experiments. Other experimenters, including

Vauclain and Halsey, Professor Benjamin, and other members of the Society, have done a large amount of work along these lines, which has been of great value to the designer as well as the user of machine tools.

6 Now, with the advent of the high speed-steel more such experiments are required, and there is a very promising field for some of the research laboratories to take up such problems as have been indicated in this paper, and ascertain how much power is required to operate such machines, and how much may be saved by cutters of different construction, tools of different shapes, and different proportions of feed and rate of cutting, and various other matters of value to the engineer.

The Author I would like to go over briefly some of the remarks which struck me most vividly. Mr. Miller refers to the milling-machine as a machine especially adapted for tool-room and jobbing work. That has been true almost to the exclusion of everything else. It is true to a very large extent even at the present time, but there seems to be a tendency to use the knee and column type of milling-machine for tool work and jobbing work, for work requiring the fine adjustment to which the milling-machine lends itself, and for work requiring the peculiar feature of the milling-machine, of lending itself to almost any kind of shape one wants to produce; but at the same time the milling machine is now being used for heavy every-day rough shaping work, plain slabbing, surfacing, etc.

2 One reason why this has not been done in the past may have been the nature of the cutters. It may be that the reason why the milling-machines were used so long almost exclusively for tool-room work was a historical one, but we seem to be rapidly drifting away from this condition; and I thought it would be well, in recognition of this fact, to have some knowledge of the milling-machine outside of its old scope; knowledge of its ability to remove chips. This is not minimizing at all the importance of the milling-machine as a tool room machine, and I realize fully that the milling-machine may be highly efficient as a machine-tool for a great many lines of work, though it may not be efficient as a user of power. But the milling-machine is also being used more and more for heavy work, and will be used in the future to an even greater extent; and in heavy work an economical user of power is in my opinion very important.

3 I wish to refer also to what Mr. Flather said, that it is not the power that cuts the figure. I would suggest, that we use just

as much power as is necessary, and not a bit more, in the machine, and take the rest of the power to drive a fan somewhere in the open air; use the power if you wish to, but do not use it in the machine.

4 The tests which I have described were carried out in order to determine what power is actually required in the machine, and I want to say here, that I realize that these tests are in no sense complete. They have not fully accomplished any of the aims I have set for myself—not even a large percentage; by far the greater part is left undone. It might, perhaps, have been well to postpone the writing of this paper until more complete data were at hand, but the mere fact of bringing this matter up before the Society may assist in starting other people, perhaps better equipped for carrying out such tests, along the same lines, or if they have been working along these lines, may lead them to publish the results of their tests.

5 Mr. Miller further refers to the relation between the power required for the drive and for feed, and objects more or less to my statement that the pressure against the cutter is practically the pressure against the table. I mentioned this, but with certain limitations, and it may be that the limitations were not put clearly enough. All the cuts taken during these tests were what might be called flat cuts; relatively wide and of little depth; the depth of the cut as compared to the diameter of the cutter was small, and for that reason the upthrust was not very large; and even under these conditions it is not quite true that the thrust against the cutter is the same as that against the table. This is not mentioned in my paper as a fact, but merely as the supporting argument which led up to my realizing the desirability of making some tests on feed efficiencies.

6 I appreciate the remarks made by Mr. Wilfred Lewis in regard to the heavy chips taken by his heavy cutters. I have had some connection with concerns making heavy machines, and have previously spent some time in shops where these heavy machines were being run, on steel, and taking extremely heavy chips. A heavy chip will make me walk around several squares; but the machines I had to deal with would not allow taking such a chip as can be made with a 165-h.p. motor. However, though the tests described do not give any information as to the efficiency of the larger machines, the horse power used on all knee and column type machines is, I believe, very much larger than on all of the heavier type of machines on which Mr. Lewis made his tests, and for that reason I believe I do not need to apologize for having limited my paper to the smaller machines. Perhaps the size is not there, but the quantity certainly is.

7 As to the necessity of regrinding the cutter after each series of tests, I may have given the impression that this was necessary, because the cutter was pulled off after each series of cuts. This was not the case, however. The cutter was reground merely to start every series of tests under the same conditions, and to make the tests on the different machines as nearly uniform as possible.

8 Referring to Mr. Taylor's desire to have causes given as well as effects, in other words to have it shown why one milling-machine should be more efficient than the other, I wish to refer him to the title of my paper, which is not "Efficiency of Milling-Machines" but "Efficiency-tests of Milling Machines:" it is not therefore supposed to give a clear account of everything pertaining to the efficiency of milling-machines, but merely an account of tests made, the methods employed and some of the results obtained; with here and there a guess as to the possible cause. I realize that a complete treatise on the efficiency of milling-machines would be of great interest, but confess that I have not sufficient data for even the foundation of such a treatise. If all the different makes of milling-machines had been tested and one particular make found superior in efficiency, and if this fact had been mentioned in my paper, some statements about the probable cause certainly would have been in order: only four machines have been tested, however, and the result can be of interest only in so far as it shows that a difference in efficiency exists.

9 I believe this paper meets the requirements set up by Mr. Taylor, namely, that it should be of some educational value to the members of the Society, though I realize that it is that only to a very limited extent. Yet the aim of the paper was, by bringing forward methods employed in testing milling-machines, to enable others to work along the same lines, and to whatever small extent this aim has been accomplished, to that extent my paper must have educational value.

10 I am very much puzzled by Mr. Taylor's remark that my statement as to the new milling-cutter has very much the appearance of an advertisement. It would seem to me that to mention that a milling-cutter exists, but not to describe it, nor to show a picture, nor to say who makes it, nor even that it is being made or ever will be made, lacks about all the essentials of a live advertisement.

No. 1216

THE DEVELOPMENT OF A HIGH SPEED MILLING CUTTER, WITH INSERTED BLADES, FOR HIGH POWERED MILLING MACHINES

By Wilfred Lewis, Philadelphia, Pa.
Member of the Society

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It has long been recognized that the process of milling is superior to any other for machining metal because the metal can be removed at a much higher rate of speed than by any other method, and as the operation of cutting is continuous, milling cutters being made up of a multiplicity of cutting edges, work can be machined at a much lower time cost. After the advent of high speed steel, builders of milling machines, especially those of the planer type, began to offer high-powered milling machines, and today it is a very common thing to see a milling machine whose spindle is driven by an individual motor of from 50 to 75 h.p. capacity, whose platen is driven by an independent motor of from 7 to 15 h.p. capacity and whose cross-rail is raised and lowered by an individual motor of from 3 to 5 h.p. capacity. While the milling machine has been developed to high power and high speed, the milling cutter has not advanced as rapidly; the user is thus confronted with a very unsatisfactory condition, the output of his milling machine being limited to cutters of inadequate capacity. This condition results from faults of design lying in the shape of the blade and method of fastening. This criticism applies to the inserted blade type of cutter, which by reason of its cheapness in first cost and maintenance has been universally adopted for heavy slab milling.

2 In 1892 George Brechtol developed a milling cutter of the inserted-blade type, in which the blades were made of carbon steel, properly shaped and secured. The results obtained at the time were considered extraordinary, being far in advance of those obtained

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

by other types of carbon steel cutters. His cutter consisted of a malleable iron core, bored and keyseated, into whose body were planed 8 helical dove-tail grooves, considerably wider than the blades. The blades were bent around a cylinder to the desired helix. They were set inside a cylinder and properly spaced by means of blocks; the core was then set in position and soft metal poured, filling the dove-tail grooves and the spaces between the blades.

3 Mr. C. D. Peck was the first to develop successfully a milling cutter of the inserted-blade type for heavy slab milling, having high speed steel blades, helical in shape. The paramount feature in Mr. Peck's cutter was the helical shape of the blades. He planed helical slots, rectangular in shape, in a steel housing, inserted therein high-speed steel blades bent to fit the helix of the slot, and held them in a rigid position by means of wedges inserted at intervals between the front face of the blades and the side of the slot, filling the spaces between the wedges with soft metal. Mr. Peck designed and built a hand press for bending the blades, which consisted of a pair of lateral jaws encircling a shaft; one of them stationary, the other actuated by a lever and toggles so as to close upon the blade to form a helix.

4 His first cutter was built at the Pittsburg works of the American Locomotive Company in 1905, and from tests made at the time, the results were far in excess of those obtained from other types of milling cutters with inserted blades, both in material removed in a specified time and in power consumed per cubic inch of material removed. He proceeded to build more cutters and put them into use, having conclusively demonstrated by continuous operation that the capacity of these cutters for removing metal was not only far greater than that of other types of inserted blade cutters, but in excess of the capacity of the modern type of high-powered milling machine.

5 Early in 1907 Mr. Peck had a conference with the manufacturers of the Taylor-Newbold Cold Saw, who were developing a milling cutter of the inserted-blade type, and the conference resulted in Mr. Peck turning his developments over to the Taylor-Newbold people. It was from this combination that the milling cutter about to be described was evolved. See Fig. 1.

6 Our primary investigations showed that there was no existing standard, or suitable rule, governing the construction of milling cutters with inserted blades, nor was there any record of exhaustive tests made to determine the most effective pitch, proper clearance

angles, or front slope and lip angles to be employed, and judging from past and present constructions in milling cutters of the inserted-blade type, the functional elements are merely arbitrary selections to suit individual tastes.

7 We also found the prevailing practice in constructing milling cutters with inserted blades to consist of cutting rectangular slots in a cylindrical housing, the slots lying in a plane angular to the axis. The angle from the axis at which the plane is set varies from 5 to 25 deg. according to the length and diameter of the housing. The blades are straight pieces of high speed steel, which are ground off after being inserted in the slots until a definite projection from the housing along the cutting edge is attained, after which clearance is obtained by "backing off."

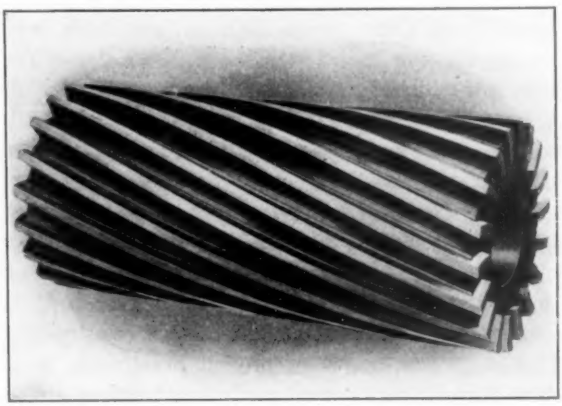

FIG. 1 HIGH SPEED INSERTED STEEL HELICAL BLADE MILLING CUTTER, 8 IN DIAMETER, 3½ IN. BORE, 18 BLADES

8 The blades are held in position by various methods, such as by clamps placed between each alternate pair of blades and drawn inward by screws; or by cutting grooves in the abutment between each alternate pair of blades and forcing the parted abutments by means of taper screws to clamp the blades on either side; or by driving wedges between the front face of the blade and the side of the slot; and by numerous other means.

9 We also found that much can be learned and applied to milling cutter blades from the development of the lathe and tool, especially the round nose tool; a fact totally ignored by the manufacturers

of milling cutters, who still try to accomplish the impossible,—that is, setting a straight blade in a slot with its front face in a plane angular to an axis, to develop thereby and maintain throughout its length the proper front slope and lip angle, and define a helix on the line of the cutting edge.

10 The first point which we considered in constructing the cutter to be described was the shape of the blade, with the following conclusions: To maintain a prescribed slope and lip angle throughout its entire length, the blade must be bent to form a helix, and by so shaping it all angles and the contour of both blades and slot would be constant over their entire lengths. Again, with the blades helical in shape, a continuous cutting edge with a constant lip angle would be maintained throughout any length of cutter.

11 The second point for consideration was the pitch or lead of the blade, and from experiments with various leads we found an effective angle for the helix to be about 20 deg. To facilitate computation, we adopted the formula, diameter $\times$ 9 = pitch, which would develop 19 deg., 15 min. as the angle of the helix.

12 The third point was the form of the grooves in the cutter blank. These had previously been planed approximately rectangular in section with a slight amount of undercutting to hold the blade and the wedges used for fastening it in place. It occurred to one of us however that this grooving of the cutter blank could be done better and faster by milling than by planing, and that an undercut groove might be produced at once by a saw set in a certain relation to the cutter blank. This suggestion was soon proved practicable, and although the groove so formed was not so easily fitted with a cutter blade on account of its curved sides, the curved sides gave the cutter a lip angle which was of great value in actual service. To form the blades accurately to the shape of the groove, it was necessary to design a bending machine of great power, capable of squeezing the blades at once to proper form not only as helices of correct pitch, but of correct curvature in a direction normal to the helix. See Fig. 2.

13 This machine was made to act in one way like the original hand-bending machine made by Mr. Peck, but in addition to the lateral jaws which closed on the blade to form the helix, it was provided with a cap which snapped quickly upon the lateral jaws, completely enclosing the blade that was being bent to proper form. Time is such an important element in the handling of high-speed steel that the value of this bending machine, which was quick enough to act before the temperature of the steel had fallen below the working point, can hardly be over-estimated.

14 Special furnaces for treating our blades were designed and built under the Taylor-White patents, but we are not at present so much concerned with the apparatus for manufacture as with the results obtained.

15 The fourth point for consideration was the method of securing the blades. Mr. Peck's experiments showed metal wedges to be neither satisfactory or economical as a means of securing the blades. While installed with ease they were exceptionally hard to remove when necessary to replace blades, owing to their tendency to imbed themselves in the housing. Furthermore, by driving them in at intervals along the high speed steel blade intermittent strains were

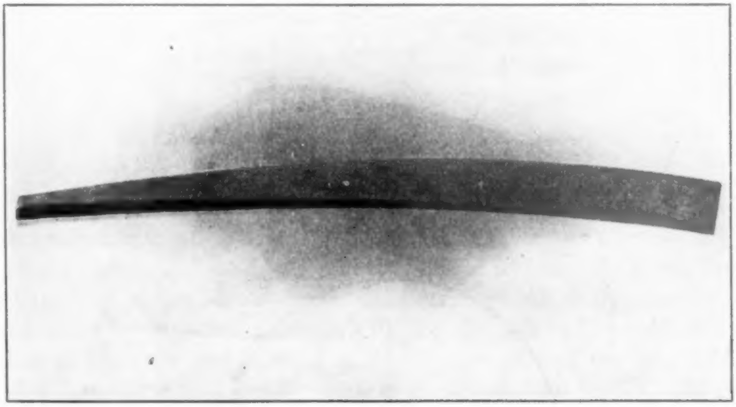

Fig. 2 High Speed Steel Blade Bent to Helical Shape and Treated by Taylor-White Process Before Insertion in Housing

developed, causing the blade to crack and spall off at the point of contact between the wedge and blade when under cutting pressure. Other mechanical fastenings were debarred either by excessive cost or by inability to withstand vibration and remain rigid.

16 Experiments were made with various alloys until a proper combination was obtained, capable of flowing freely, cooling without shrinkage, withstanding great strains without crumbling and being removed quickly and economically.

17 A device was designed for compressing the alloy in the slots after it had been poured, and a device for removing the alloy when replacement of blades is necessary. With the alloy compressed in the slots we are able to secure the blades in an anchorage sufficiently

rigid so that the blades may be broken off by sheer force without affecting it. This form of construction enables us to produce a cutter of moderate diameter free from torsional strains and with a greater number of blades for a given diameter, and possessing a capacity in excess of the requirements of high-powered milling machines.

18 The milling cutter in question will be referred to here as a "helical" rather than a "spiral" cutter. The prevailing practice is to use these terms synonymously, but geometrically they are not even analogous; a helix is a line generated by progressive rotation

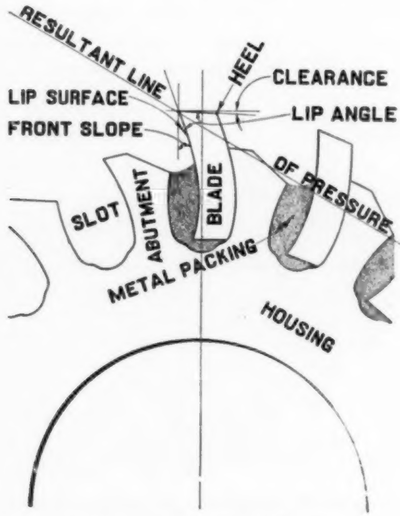

FIG. 3 DESIGNATION OF CUTTER PARTS

of a point around an axis and equidistant from its center, while a spiral is generated by progressive rotation of a point around a fixed axis with a constantly increasing distance from the axis.

19 Fig. 3 shows the meaning of the following terms: housing-body of cutter; abutment—support of blade; metal packing—anchorage; slot; blade; heel of blade; clearance angle; lip angle; front slope; lip surface and resultant line of pressure. The side slope is defined by the angle of the helix.

20 Fig. 4 shows how the chip is partly torn and partly sheared from the body of the forging, the cutting edge of the blade not being under heavy pressure. Our experiments have shown conclusively that the closer the center of pressure of the chip is to the cutting edge, the

greater its intensity, and the generated heat is largely concentrated towards the cutting edge, where the sectional area to carry it off is much less. Then again the frictional heat generated by lack of back slope becomes so great as to cause cohesion between the chip and the cutting edge. Under heavy pressure we have seen a compact of chip and blade which was virtually inextricable, so that when disunited from the blade the cutting edge still adhered to the chip. To eliminate

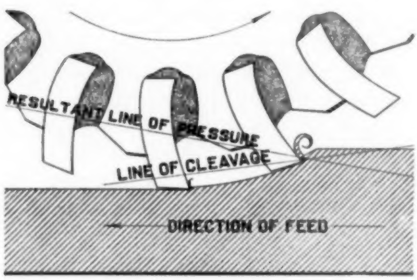

FIG. 4 ACTION IN REMOVING CHIP

this condition where straight blades are employed, the practice is to set the front face of the blade slightly back of a radial plane (see line R, Fig. 5) to assure a front slope. It is well known that the absence of front slope eliminates any proclivity which the lip may have to develop a line of cleavage and thereby throw the pressure of the chip back on the lip surface from the cutting edge.

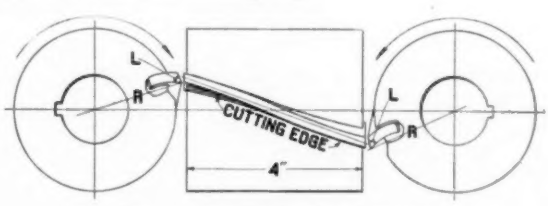

FIG. 5 CONSTANT LIP ANGLE

21 Fig. 5 shows a constant lip angle L throughout the entire length of the blade, which is set at an angle of 20 deg. to an axial plane, this angle remaining constant throughout any length by reason of the blade's curvature.

22 Fig. 6 shows a varying lip angle L from maximum at line R1 to minimum at line R2 in a straight blade set at an angle of 20 deg. to an axial plane. This condition limits the length of blades.

23 In Fig. 7 a straight blade is set in a plane radiating from the axis, designated by line R1, and by carrying it across the face of the housing at an angle set at 20 deg. to an axial plane. You may observe the development from no front slope to a positive front slope, designated by the letter S. In milling, a blade with this irregularity in the front slope causes the cutter to drag on one side and gouge on the other. Blades of this type cause excessive vibration to the cutter, due to the varying angle of the front slope, and necessarily consume more power. It has been contended that by nicking the blades to alleviate

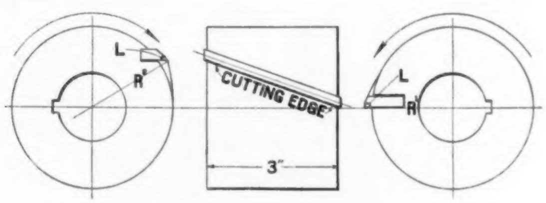

Fig. 6 Varying Lip Angle

the tendency of the blade to gouge and drag, and to define a more even pressure throughout the line of the cutting edge, less power would be required to drive the cutter; the assumption being that less power would be consumed in breaking up the chip than would be required by a continuous cutting edge producing a continuous chip.

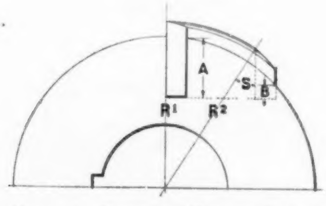

Fig. 7 Varying Front Slope

24 We differ with this contention, having demonstrated conclusively by experiments that the initial fault is caused by absence of front slope, and that an undesirable feature is developed by nicking the blades. The blade behind the nick whose cutting edge covers the gap formed by a nick in the blade preceding must accept double the feed; this causes chatter and produces an uneven machined surface.

25 It is also a fact that when a rectangular slot is cut in the face of a housing at an angle of 20 deg. to an axial plane, its depth becomes gradually less as it progresses across the face, until it reaches a vanishing point designated by lines A and B in Fig. 7. This condition not only limits the width of the cutter and necessitates a housing of large diameter when a medium width of cutter is required, but develops an anchorage possessing the required strength and rigidity at one side and decreasing in proportion to the length of blades at the other. Where a wide face is required it is necessary to construct a cutter composed of a number of sections.

26 Fig. 8 illustrates a cutter with straight inserted blades made up in sections, each alternate blade overlapping the blades in the opposite section, so as to obtain the desired width of face. The sections are so set that the cutting edge forms a continuous line. Above the cutter is a diagram showing the relation of the blades in one section to those in the other.

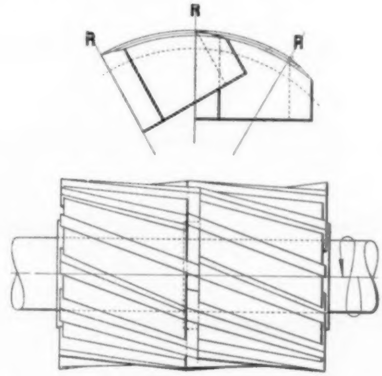

FIG. 8 OVERLAPPING INSERTED BLADES

27 Too much stress cannot be laid on the use of a lubricant during the process of milling. A copious stream of lubricant falling at slow velocity should be thrown directly upon the chip at the point of removal. Heat generated by the pressure of the chip is the chief cause for wear, and if allowed to become too great it will soften the lip surface of the blades and cause them to crumble or spall off. An ample supply of lubricant during the milling operation carries off the heat, materially lessening the dulling of the cutting edges.

28 From our experiments and those of others it has been conclusively shown that a gain of 33 per cent in the cutting speed in milling

steel and wrought iron is made by throwing a heavy stream of lubricant upon the cutter and along its entire face; and a gain of 15 per cent in milling cast iron. The piping for conveying the lubricant to the milling cutter should be arranged with nozzles spaced about 4 in.

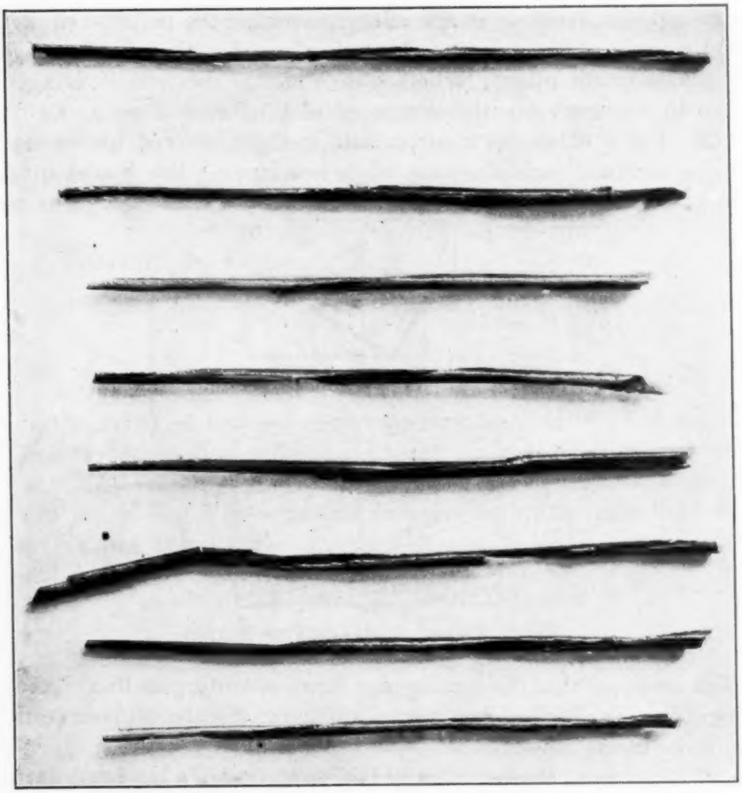

Fig. 9 Steel Chips Milled From 0.57 Per Cent Carbon Steel Forging 18 in. Wide at the Rate of $47\frac{1}{4}$ cu. in. Per Min.

These chips vary from 14 in. to $16\frac{1}{2}$ in. in length

apart, in sufficient number to cover the face of the cutter. The main supply pipe to the nozzles should be large enough to supply each nozzle with from two to three gallons of lubricant per minute. The general arrangement, size of supply pipe and number of nozzles will

be regulated by the width of the machine and the nature of the work to be done.

29 In the more recent designs of slab milling machines due consideration has been given to lubrication. The platen is drained by

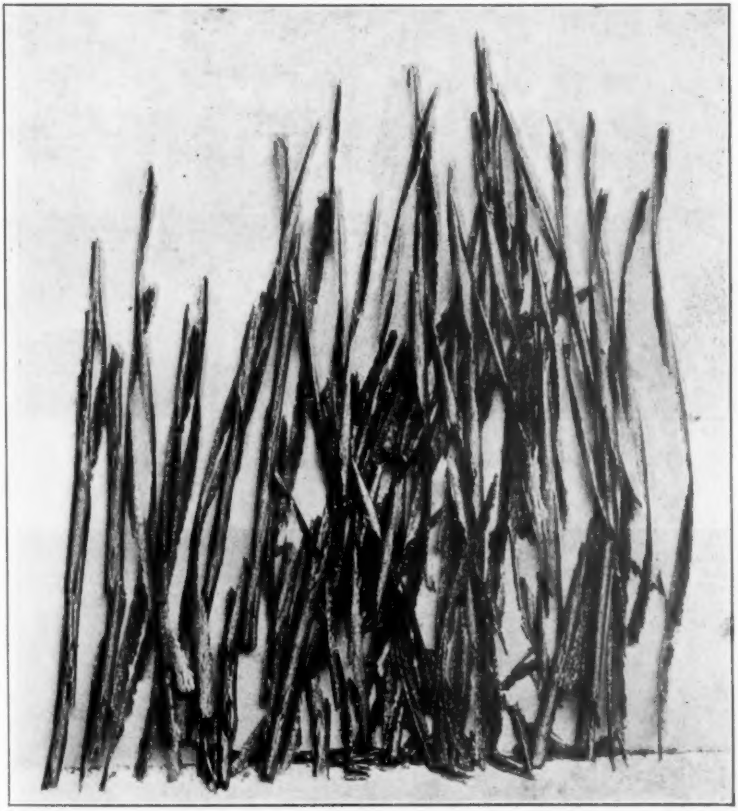

FIG. 10 CAST IRON CHIPS MILLED FROM A SURFACE 14 IN. WIDE.

THESE CHIPS WERE REMOVED AT THE RATE OF 105 CU. IN. PER MIN. AND AS A WHOLE ARE STRAIGHT AND CYLINDRICAL IN SHAPE. THEY AVERAGE ABOUT 12 IN. IN LENGTH AND ARE STRONG ENOUGH TO SUPPORT THEMSELVES WHEN STANDING ON END.

gravity to a tank located at the side of the machine, from which the lubricant is raised by a pump to a reservoir formed by the cross piece at the top of the housings, to which is attached the nozzle bracket that conveys the lubricant to the required point of gravity.

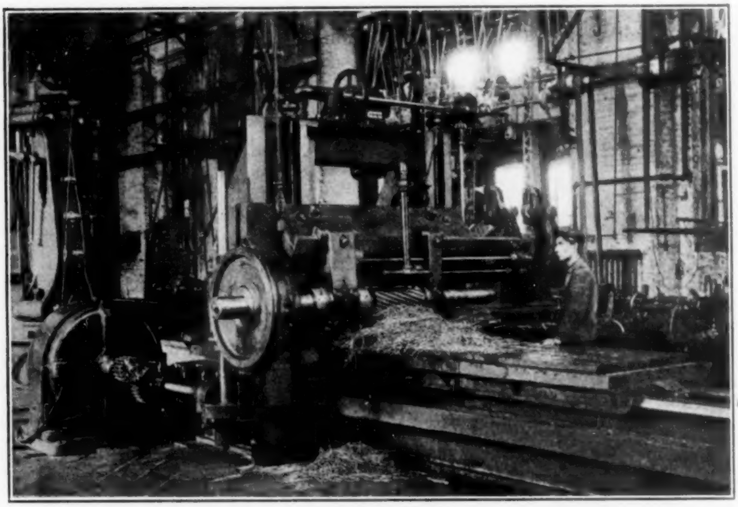

Fig. 11 Bement-Miles High-Powered Milling Machine

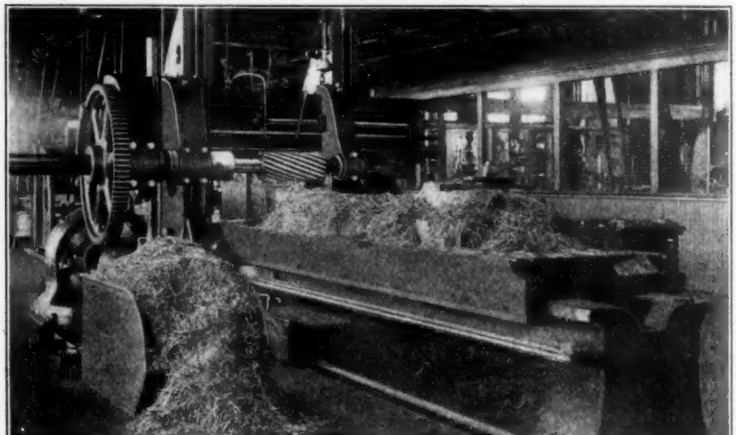

Fig. 12 Machine at Completion of Cut

30 In the tabulated tests the amount of power consumed by the milling machine will be given as read from the voltmeter and ammeter. Dynamometer readings of the actual power exerted at the spindle, or the power consumed by friction in the gearings and bearings, will not be given consideration; the user being most interested in the commercial readings that indicate the power for which he has to pay.

31 Credit is due the Bement-Miles Works of the Niles-Bement-Pond Company, of Philadelphia, for the use of various types of high-powered milling machines, and valuable assistance in carrying out experiments and tests.

32 The authors regret that in order to present the above data at the annual meeting of 1908 it has been impossible to complete experiments.

TABLE 1 SLAB MILLING CAST IRON, TAYLOR-NEWBOLD HIGH-SPEED STEEL MILLING CUTTER, 8-IN. DIAMETER, 18-IN. FACE 18 INSERTED BLADES. TEST MADE AT BEMENT-MILES WORKS, JULY 13, 1908

MACHINE USED: 42-IN. BEMENT-MILES MILLING MACHINE
DRIVING MOTOR: WESTINGHOUSE DIRECT-CURRENT CONSTANT SPEED TYPE 40-H.P. AT 220 VOLTS, 153 AMPERES
MATERIAL CUT: CAST IRON TEST BLOCK, 15 IN. WIDE, 36 IN. LONG

Number of test	CUT							Duration of test	SPEED OF CUTTER		ELECTRICAL READINGS			H.p. per cubic inch removed per min.	Cu. in. removed per min. per inch width of cut
	FEED		Depth Inches	Width Inches	MATERIAL REMOVED				R.p.m.	Feet per minute	DRIVING MOTOR				
	Table advance per minute Inches	Advance per blade Inches			Cubic Inches per minute	Pounds per minute	Pounds per hour				Amperes	Volts	H.p.		
1	3½	.00636	½	15	26.25	6.83	410.13	10m. 2s.	25	53½	150	200	40.21	.53	1.75
2	5⅞	.01332	½	15	44.06	11.47	688.38	6m. 8s.	24½	51⅓	150	220	44.23	1.00	2.94
3	7½	.01736	½	15	56.25	14.74	737.33	4m. 48s.	24	50¼	160	208	44.70	0.794	3.75
4	6⅛	.01361	½	15	45.93	11.96	717.72	5m. 5s.	25	53½	175	208	48.79	1.060	3.06
5	7¾	.01722	½	15	58.12	15.13	908.14	4m. 39s.	25	53½	225	200	60.32	1.030	3.87
6	7⅝	.01781	11/16	15	50.70	13.20	792.13	4m. 53s.	23	48	260	204	70.96	1.390	3.38
7	7⅝	.01781	11/16	15	50.70	13.20	792.13	3m. 39s.	23	48					
8	8	.01851	11/16	15	82.50	21.48	1288.98	1m. 15s.	24	50¼	240	213	68.52	.830	5.50
9	7	.01760	1	15	105.00	27.34	1640.52	3m. 17s.	22	47	350	190	89.14	.849	7.00

TABLE 2 SLAB MILLING STEEL, TAYLOR-NEWBOLD HIGH-SPEED STEEL MILLING CUTTER, 8-IN. DIAMETER, 18-IN. FACE, 18 INSERTED BLADES. TEST MADE AT BEMENT-MILES WORKS, JULY 14, 1908

Machine Used: 42-in. Bement-Miles Milling Machine
Driving Motor: Westinghouse Direct-current Constant Speed Type 40 h.p. at 220 Volts, 153 Amperes
Material Cut: 30 per cent Carbon Steel Test Block 18 in. Wide, 20 in. Long

Number of test	Cut							Duration of test	Speed of Cutter		Electrical Readings			H.p. per cubic inch removed per min.	Cu. in. removed per min. per inch width of cut
	Feed				Material removed						Driving motor				
	Table advance per minute Inches	Advance per blade Inches	Depth Inches	Width Inches	Cubic Inches per minute	Pounds per minute	Pounds per hour		R.p.m.	Feet per minute	Amperes	Volts	H.p.		
1	11¼	.01785	5/32	18	31.64	8.96	537.81	1m. 47s.	35	73⅓	190	220	56.03	1.77	1.76
2	11¼	.01785	5/32	18	63.28	17.92	1075.62	0m. 11s.	35	73⅓					3.51
3	6⅜	.01041	⅜	18	43.03	12.19	731.44	1m. 30s.	34	71	300	195	78.41	1.82	2.39
4	7	.01080	⅜	18	47.25	13.38	801.15	1m. 26s.	36	75⅓	400+	180	96.51	2.04	2.62
5	7	.00925	⅜	18	47.25	13.38	801.15	2m. 51s.	42	88	370	187	92.74	1.96	2.62

TABLE 3 MILLING CHANNELS, TAYLOR-NEWBOLD HIGH-SPEED STEEL MILLING CUTTER 4⅜-IN. FACE, 8-IN. LISTED DIAMETER, 8¼ IN ACTUAL DIAMETER, 18 IN. INSERTED BLADES, 3½-IN. BORE. TEST MADE AT BEMENT-MILES WORKS, OCTOBER 20, 1908

MACHINE USED: 42-IN. BEMENT-MILES MILLING MACHINE
DRIVING MOTOR: WESTINGHOUSE DIRECT-CURRENT CONSTANT SPEED TYPE 40-H.P. AT 220 VOLTS, 153 AMPERES
MATERIAL CUT: 35 PER CENT CARBON STEEL FORGING

Number of test	Cut: Feed: Table advance per minute Inches	Cut: Feed: Advance per blade Inches	Cut: Depth Inches	Cut: Width Inches	Cut: Material removed: Cubic Inches per minute	Cut: Material removed: Pounds per minute	Cut: Material removed: Pounds per hour	Duration of test	Speed of Cutter: R.p.m.	Speed of Cutter: Feet per minute	Electrical Readings: Driving motor: Amperes	Electrical Readings: Driving motor: Volts	Electrical Readings: Driving motor: H.p.	H.p. per cubic inch removed	Cu. in. removed per min. per inch width of cut
1									39	82.95	35	220	10.32		
2	6⅞	.01320	5/16	4⅜	9.40	2.66	159.78	3m. 25s.	37	78.69	85	200	22.74	2.42	2.14
3	1 15/16	.00307	1	4⅜	8.48	2.40	144.14	7m. 12s.	35	74.44	95	185	23.56	2.77	1.93
4	1½	.00231	1	4⅜	6.56	1.86	111.51	4m.	36	76.57	85	193	21.85	3.33	1.49
5	2⅝	.00394	1 5/16	4⅜	15.07	4.27	256.15	2m. 40s.	37	78.69	113	200	30.29	2.00	3.44
6	3⅝	.00544	1 5/16	4⅜	20.81	5.89	353.72	1m. 56s.	37	78.69	135	195	35.28	1.69	4.75
7	5	.00750	1 5/16	4⅜	28.71	8.13	488.01	1m. 24s.	37	78.69	200	190	50.93	1.77	6.56
8															

DISCUSSION

MR. FRED J. MILLER Most of us are, of course, familiar with the fact that milling cutters were first used generally for producing forms otherwise difficult to produce at reasonable cost. Parts of small arms and similar things were thus produced as the next step in advance of filing them to the required form and dimensions. Cutting the teeth of gear wheels was one of the first if not the very earliest use of the milling cutter.

2 In all such work the form of the cutter and the ability to keep it near its original form until worn out were the chief considerations; as indeed they still are for many kinds of work. This, more than anything else, probably, has led to the almost universal use of the radial front face for the teeth of milling cutters; because only by having radial front faces could the problems connected with forming cutters and maintaining their forms be sufficiently simplified to be at all practicable. This seems to have led to what may be called the traditional practice of making the front faces radial in all cutters —even in the plain, solid, spiral cutters whose only office is to remove metal from a plane surface and which can as well be made and sharpened if they have front rake as with radial surfaces for the front of the teeth.

3 Radial surfaced teeth can scarcely be said to cut. They push and jam the metal off. When heavy cuts are taken, this jamming becomes a serious matter and front rake is almost a necessity. Even a little front rake is very advantageous when the mere removal of metal from a plane surface is the object sought and it seems likely that such rake will become as common for milling cutters used on such work as it now is for lathe and planer tools and for the same reason.

MR. OBERLIN SMITH I want to ask the authors of this paper what they found to be the best lubricants (or "coolicants") for cast-iron, for mild steel and for brass respectively, also the best angle of rake, that is the angle with the radial line, in lathe-work, etc., respectively for the three metals mentioned.

2 Changing the subject a little, I ask the membership in general if they have had any experience either with boring toolsor milling-cutters made by inserting a set of blades into grooves in wooden patterns, and molding them so that they remain in the sand for the casting of a hub or body about them, the same being hollow to go on a

boring-bar or having a shank of its own to insert in a milling-machine spindle. I had some experience years ago in making such tools of from three to nine inches in diameter of mushet steel, finished in a grinding-lathe. It is not practicable to use carbon steel, as heating it for hardening is apt to crack the iron hubs. No doubt this scheme has been tried by a good many people, but with what success?

3 Doubtless high-speed steel cutters could be cast in this way, provided the heat of the cast-iron would not anneal them too much. That is a point to ask Mr. Taylor or Mr. Lewis about, or any one who has found out anything about it. Is there any practical way of using high-speed cutters, and casting them into a hub of iron? It is obvious that a large-toothed cutter can be made more cheaply in this way than in any other, but we do not want the slow speeds of mushet steel.

Mr. Fred. W. Taylor The writers of this paper, as well as that of Mr. DeLeeuw, speak of the "lubricant" used upon the tool and Mr. Oberlin Smith has just spoken of the best "lubricant" to use on a tool. In taking a heavy cut with a tool, in fact in doing any metal cutting in which a chip runs continuously across the lip-surface of a tool (except in the case of a light finishing cut) is it possible to get any lubricant between the chip and the tool? In cutting steel the tool receives a pressure from the chip of about 180 000 lb. per square inch at the spot where the chip rubs upon it. Is it possible to get any lubricant between two surfaces, one of which is forced against the other with a pressure of 180 000 lb. per square inch on an average, and in which one surface is continuously moving past the other at the rate, we will say, of 40, 50, 60 or 70 ft. a minute?

2 I think this is utterly impossible when the nose of a tool is buried in the steel as it is in taking a roughing cut. The word "lubricant" is a survival from the practice of pouring a light trickling stream of water or oil on a finishing tool from a small water can supported above the tool, the function of the water or oil being to produce a polishing or burnishing effect upon the work. Water was not used in this way to cool the tool and was confined to finishing tools. Doubtless under a light finishing or scraping cut at the last instant a small amount of water or oil does find its way between the tool and the work.

3 On heavy roughing cuts, it is impossible to get any lubricants between the chip and the tool surface. It is, however, possible and

desirable, in almost all roughing cuts, to have either water or oil thrown upon the chip and the tool for the purpose of cooling them, since, as pointed out by the writer in his paper, On the Art of Cutting Metals, a tool cooled in this way can cut from 15 to 40 per cent faster than a tool running dry. Cold water is the best conductor of heat, better than any of the oils, therefore it is the best of the cheap cooling-mediums to throw onto a tool. The only object of putting soap in the water on heavy cuts, or of putting soda in the water, is to stop rusting on the machine or work when the water splashes over. Cold water is the proper thing to pour on a milling-machine of this type. It cools better than any other cheap material known, and the cooler it is the better. The use of soda in water, or the substitution of oil for water on roughing cuts, is merely to stop rusting.

Mr. A. L. DeLeeuw Mr. Taylor's remarks upon lubricants brings to mind an experience I had just about one year before the famous Taylor-White steel was brought out. High speed for tools was in the air, and I tried to get results which have since been accomplished in an entirely different way, by simply cooling the tool and the chip. Realizing that it was impossible to force a lubricant of any kind between the tool and the chip, and that at the same time forcing the lubricant somewhere else would not cool the cutting point of the tool sufficiently to keep it from burning out, I had a ring constructed and attached to the compressed air supply. The ring was provided with a number of small holes focusing at a common center and so adjusted on the lathe, as to bring the focus of all these small streams of air some little distance from the tool point. The air expanded, forming a center of refrigeration easily determined by a thermometer or by the finger. By adjusting the ring in such way that the center of refrigeration coincided with the cutting point of the tool, it was possible—and that was a year before the high-speed steels were brought out—to turn cast iron which had been rejected on account of its hardness, at the rate of 168 ft. a minute, and I do not know how much faster, because the lathe would not pull it. It was even possible to run steel at the rate of 250 ft. a minute, and I do not know how much faster it might have been run, if the lathe had allowed it.

2 This brings out strongly the same point Mr. Taylor has made, that the lubricant simply cools the tool. There is no object in cooling the chip, but of course we cool the tool by keeping the chip cool.

Mr. Oberlin Smith I want to go the gentleman who has just spoken one better. Some years ago, before high-speed steels came up,

I came to the conclusion that all that is wanted is to cool the tool. The coolest thing I could get was liquid air. Procuring a can containing 15 gal., and taking every precaution to keep it in a liquid condition, I rigged up on a lathe an ordinary two-quart can with a ¼ in. pipe and spigot, and filled it with liquid air. I covered it with flannels, and tried to get some of it on the tool, but it would not run. The reason of course was that it expanded so fast in the small pipe that as a gas it pushed the liquid back into the can and held it there. The pressure upon the atmosphere was so violent that it held against a head of several inches. I then put in a ⅜-in. pipe, but it would not run. With a ½-in. pipe it did run but of course my supply did not last very long.

2 I have not any accurate records of just what speeds were obtained, but I did get a great deal higher speed than with usual ordinary carbon steel. With a one-inch drill the speed could be doubled in cast-iron. The liquid did not get well down among the chips, but if it had been forced down even greater results might have been obtained. If liquid air were properly handled and forced against the tools it would probably be the best cooling-agent we could secure. This, however, would seem wholly impracticable on the score of expense. If it could be forced immediately to the machines it might be effective, but in the ordinary machine-shop we start and stop the tools frequently, and where the pipes would have to be kept well insulated to keep the liquid from getting warm it doubtless would not be feasible.

3 I had thought of patenting the combination of liquid air with a lathe milling-machine, etc., but Mr. Taylor's high-speed steel came out and cooled my enthusiasm below the temperature of the air. I now hereby give the invention freely to the world—if I am the original inventor of it—because I do not know whether it is really good for anything, and it is too much trouble to find out.

4 It is an open question whether steel gets brittle at low temperatures and would become so if cooled by liquid air. The metal being cut, and the tool itself, would probably become heated enough, however, so that there would not be this effect.

Mr. A. B. Carhart The subjects of lubricants may be a little apart from the main purpose of the paper, but the subject should not be allowed to drop where it is, after the remarks of Mr. Taylor. Why have great advantages been claimed for many years for lard oil and its compounds for lubrication in milling machine operations?

The advantages of kerosene mixtures are recognized for cutting aluminum, bronze castings and such metals. The greater conductivity of the lighter oil is easily recognized, but is there no other reasonable explanation of the predilection in favor of kerosene over the other oils; and aside from the rusting tendencies of clear water are there no reasons why mixtures of water with various skim milk compounds offered as substitutes for lard oil have any real value? If there are no advantages in oil other than its non-corrosive qualities, why is it that so much of the cutting oil on the market is so strongly corrosive of the machine tool members?

PROF. R. T. STEWART In regard to lubrication under high pressures, there are industries in which that is effective, for example, in the cold-drawing of seamless steel tubes; if you attempted to draw steel tubes without lubrication, the effort would be futile. If you want a reduction in cross-section of say 5 to 10 per cent, you may use one lubricant successfully; but if you wish to get a reduction in cross-section of say 25 or 30 per cent, by cold drawing, you have to use another lubricant, one that is better adapted to the purpose. I have effected reductions in cross-sections of 33⅓ per cent, by cold-drawing the tube through a die and over a mandril, when using a proper lubricant.

2 I should not be surprised, though I have had no considerable experience in using lubricants with tools, if even there they had an effect. I do not know what the surface pressure is in drawing a seamless steel tube, but it must be fully as great as the pressure required to lift the chip in tool-cutting. Seamless tubes have been drawn of high carbon steel, and I should think in that case the pressure would be in the neighborhood of 150 000 lb. per sq. in., at least, and the tubes were drawn quite successfully by the use of proper lubricants. Without the use of lubricants, they could not be drawn.

MR. FRED. W. TAYLOR It appears to me that the two cases are not in any way parallel. In the tube-drawing the lubricant can be put on the tube before it starts into the die and a certain amount of it will remain upon it while it passes through it. But how is it possible to make a lubricant run uphill to the nose of a roughing-tool which is ploughing its way into a forging and is completely buried at all times beneath the chip? It would be necessary for the lubricant to force its way uphill between two surfaces under 180 000 lb.

pressure. The nose of the tool is buried at all times beneath the chip and the chip travels down on top of the tool continuously at the rate, say of 60 ft. per minute. It would be quite as impossible in the case of tube-drawing to force the oil up through the bottom or rear end of the die between the die and the tube.

Prof. R. T. Stewart I believe it is the practice in milling to apply the lubricant so that it comes in contact with the cutter before it enters the metal, and some of it will surely cling to the cutter, just as in drawing seamless tubes. I had in mind the case illustrated in the paper, which is upon milling cutters. In milling it is always possible to apply the lubricant to the cutter.

Mr. Fred. W. Taylor In my preceding remarks I had in mind a lathe tool. In the case of a milling-cutter I stand corrected by Professor Stewart—it is possible to get a lubricant onto the lip surface of a milling-cutter before it starts into its cut and in doing fine finishing-work with a milling-cutter a lubricant is frequently desirable. The large milling-cutter under discussion is for taking heavy cuts and in its case the lubricant is of no use. In using this cutter properly an enormous stream of water is thrown on the blades for the purpose of cooling them so as to get a higher cutting-speed.

2 Answering Mr. Smith's question as to the possibility of pouring cast-iron around a high-speed blade, if you treat the blade so as to give it the best high-speed properties, it must be heated, say, to a high heat between 2000 and 2400 deg. fahr., and then cooled continuously to below 1200 deg. If during the process of cooling from the high heat down to below 1200 deg. it is reheated, even for a short time, the high-speed property will largely disappear, and if during this cooling operation it is reheated for the space of a few minutes, its high-speed is almost entirely lost. Suppose you give a blade its high-speed properties by heating it to a temperature of 2400 deg. and allowing it to cool, and then put this blade back into molten iron which is at a temperature, say, of 1800 deg. This reheating would largely destroy the high-speed properties of the blade. If high-speed steel is heated beyond 1250 deg. and held there for two minutes, the high-speed qualities will be almost entirely gone. And they will begin to return to the blade a second time only after a temperature of 1725 deg. has been passed. Then again, even although its high-speed properties were partially restored to it by heating it, say, to 1800 deg. the grain would be coarsened to

a great extent. If high-speed steel is held at a heat above 1800 deg. for even ten minutes it will deteriorate, and if held there for an hour it will become about as brittle as chalk.

3 For cutting brass the tool should have very little if any back-slope or side-slope. A brass-cutting tool should be almost a scraper. Its cutting-edge should not be rounded out in the least, however, or allowed to be in the least dull. Brass tools should for the most part be whetted after being ground. For phosphor-bronze the rate must be different. Tools for cutting steel as hard as fire steel or harder should have 5-deg. back-slope and 9-deg. side-slope. For cutting cast-iron and medium steel they should have 8-deg. back-slope and 14-deg. side-slope; for mild steel, 8-deg. back-slope and 22-deg. side-slope.

Prof. J. Burkitt Webb suggested in regard to the plan proposed for using liquid air as a cooling-agent that it might be as well to conduct the air through the pipes before it was liquefied, and allow it to liquefy in a spray as it came from the nozzle. It would be better than water in some respects as it would not require a drain-pipe to get rid of the waste. He asked if experiments had been made to ascertain how long oil will stick to the surface of cutting tools. He referred to the operation of his viscous dynamometer in which metal discs revolve rapidly in a case filled with water, which is constantly supplied fresh to keep it from heating unduly. The hands of experimenters had of course been in contact with oily tools, but had been wiped as clean as possible with new cotton waste; yet it was found that oil enough remained on the hands to lubricate instantly the stream of water supplying the case, and reduce the friction 10 per cent. On removing the hand, the friction would creep back to normal in a minute or two.

Mr. Fred. J. Miller As to the pressure between the cutting tool and chips, why is it when you cut steel dry the surface of the chip is rough and dry, and when you put water on it the surface which has been in contact with the cutting tool is polished?

2 We all know that in chasing threads in the lathe, especially on tool steel, lard oil will enable results in the way of a smooth surface which are not obtainable by lubricating oils or so far as I know with any other substance. If Mr. DeLeeuw noticed whether the surface of the chips which were turned from his steel at the rate of 250 ft. per min. were smooth as when a lubricant is used, or whether they were

rough as is ordinarily the case when turned dry, it would throw light upon this matter.

THE AUTHORS In presenting this paper the authors were well aware that they had hardly begun to demonstrate the possibilities of high-speed steel in a milling cutter of the type described, and their chief object in bringing it to the attention of the Society was to reap the benefit of discussion. We are very much pleased, therefore, to adopt the suggestions made by Mr. Miller in regard to the tabulation of results.

2 We want to know, of course, the maximum performance of a milling cutter per inch of face for every size of cutter made. This naturally depends upon the strength of the cutter blade when the cutter is short and on the strength of the arbor when the cutter is long. We do not yet know on what length of cutter the full strength of the driving arbor can safely be thrown, but from experiments we believe that the $3\frac{1}{2}$-in. arbor used to drive our 8-in. cutter 18 in. long would be overloaded before the cutter blades, on an evenly distributed cut.

3 We also know that the cutting speeds were very low, and that with more power and higher cutting speeds our results might easily be multiplied two or three times. The present problem is not what the cutter can do, but how it can be driven at its full capacity. The discovery of high-speed steel has immediately created a general demand for more powerful machine tools, but its adaptation to milling cutters has progressed more slowly and there are no milling machines on the market today capable of driving our cutter to its full capacity.

4 Though we do not yet know the capacity of our cutter blades per inch of length, we can estimate pretty nearly on the capacity of a driving arbor $3\frac{1}{2}$ or 4 in. in diameter and, allowing 20 000 lb. per square inch as a permissible shearing stress in a 4-in. arbor, we find it capable of driving a cut of about 60 000 lb. on the periphery of an 8-in. cutter. Our cutting speeds on steel varied from 50 to 80 ft. per minute without distressing the cutter blades at all and without any discoloration of the chips, the maximum thickness of which seldom exceeded 0.005 in.

5 In Mr. Taylor's treatise, On the Art of Cutting Metals 60 ft., is given as the proper cutting speed for a $\frac{3}{16}$-in cut, $\frac{1}{16}$-in. feed; in Paragraph 1186 it appears that the same cut can be taken with a straight cutting-edge 1 in. long at 40 per cent higher speed, or a speed of 84 ft., and in the latter case the thickness of the chip is 0.017

in. It also appears that by the use of water this cutting speed might be increased to 110 ft., all on 33-carbon steel. In our milling cutter, however, we have a shaving not one-third as heavy, and each blade cuts about 5 per cent of the running time and cools off under a stream of water during the remaining 95 per cent. The conditions are therefore particularly favorable for a high cutting-speed and it would not be at all surprising if we could take a cut of 60 000 lb. at a speed of 150 or 180 ft. a minute. This means that 300 h.p. might be consumed in milling and that with an efficiency of 75 per cent in the milling machine 400 h.p. might be required to drive it. Such enormous power concentrated on an 8-in. cutter may yet be realized, but the problem will be to design bearings that will carry 60 000 lb. at a speed of 75 or 80 r.p.m. They cannot be very long to distribute such a load properly and must be kept cool under about 1000 lb. per square inch. A machine with half this driving power would be far beyond common practice today and it is pretty safe to predict that our milling cutter will exceed in capacity any machine that may be built for some time to come, except of course when the cutters are short and the strength of the cutter blades becomes the limiting factor.

6 So much for the future development of the milling machine; but the all-important factor of interest to the user of the present type of milling machine is, how does this cutter compare with the various types of inserted blade cutters now in use. We have carried out a series of comparative tests with various types of inserted blade cutters and have found that when operated under exactly the same conditions our cutter showed a saving of 50 per cent in consumption of power for a given amount of material removed, and that the life of the cutting edge of the blades was double that of other types of cutters where straight inserted blades were employed.

7 The values of various lubricants and the effects of various lip angles on different materials are questions so thoroughly answered by Mr. F. W. Taylor that we need say nothing further.

No. 1217

METAL CUTTING TOOLS WITHOUT CLEARANCE

By James Hartness, Springfield, Vt.
Member of the Society

This paper sets forth a turning tool that is intended to cut without clearance.

2 It consists of a cutter and a holder so constructed as to allow the cutter a slight oscillatory freedom in the holder. The center line on which the cutter oscillates is substantially coincident with the cutting edge. The oscillation of the cutter about the center line does not affect the position of the edge, but it does allow the face of the cutter to swing around to conform to the face of the metal from which the chip is being severed.

3 The objects of this construction are to make possible the use of more acute cutting edges in order to reduce the cutting stresses; to equalize wholly or partly the unbalanced side pressure on the cutting edge, and to obtain a rubbing contact to prevent lateral quivering.

4 In order to bring out these objects it is necessary to analyze briefly some of the conditions under which metal is worked in a lathe, dealing particularly with cutting angles, clearance of cutting edges, and the importance of minimizing the tendency of the work and tool to separate under cutting stresses.

5 No attempt is made to discuss the forms of cutting edges for withstanding the heat of high speed service. High speed tool forms have been ably and perhaps conclusively treated in the paper by Mr. Fred. W. Taylor and its discussion, and in the papers of Dr. Nicolson before this Society and before the Manchester Association of Engineers.

6 The generally accepted cutting angle of greatest endurance under high speed is about 75 deg., and the angle of least resistance,

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

References mentioned: Dr. J. T. Nicolson's papers in Transactions of Manchester Association of Engineers, 1903, and in Transactions of this Society, p. 637, vol. 25, 1904; Mr. Fred. W. Taylor's paper, p. 31, vol. 28, 1907; also cuts on p. 333 in Dr. Nicolson's discussion of Mr. Taylor's paper.

according to some of Dr. Nicolson's tests, is about 60 deg., with an increase *below* as well as above that angle.

7 The cutting angles of the tool described in the present paper may be varied from the present orthodox angles down to 30 deg. or less, according to the nature of the work.

8 The results obtained by Dr. Nicolson, which showed an increase in cutting stress for tools more acute than 60 deg., may have been due to the cuts having been run without cutting oil or suitable cutting lubricant. Furthermore, the comparative lack of durability of the more acute edge below 70 deg. may have been due either to heat or lateral quivering, or both. The heat would have been greatly reduced by a liquid cooling medium, especially one having some suitable lubricating qualities, and the lateral quivering may now be eliminated by means explained in this paper. The thin edge of an acute tool is obviously the least suited to carry off heat or to withstand the quivering incident to cutting.

9 Having mentioned the great work of Mr. Taylor and his co-workers, and of Dr. Nicolson, it is necessary at once to disclaim any pretension to contributing valuable data, such as are found in the papers of these truly scientific researchers. Nothing of the kind is possible at this time. All that is attempted is to suggest a scheme for widening the field of investigation.

10 Instead of approaching the subject as a scientist bent on getting exact data regarding performance of certain existing forms of tools and machines, the writer's line of approach has been from the standpoint of a designer and manufacturer of lathes, and particularly lathes of the character of the flat turret lathe.

THE CLASS OF WORK HERE CONSIDERED

11 The means for cutting set forth should be considered from the standpoint of one who sees nothing but lathe work under 20 in. in diameter, and of the kind usually found in any machinery building plant, whether it is a navy yard, railroad shop, or automobile building plant; not that the means are of no value in larger work, but being out of the writer's range of experience, such work was not considered in designing the tools described.

12 A more exact description of the range of work for which this tool is intended would be: lathe and turret lathe work under 20 in. and over 4 or 5 in. in diameter, and less than 8 or 10 in. in length; also work up to 2 and 3 ft. in length, of diameters under 3 to 3½ in. and generally over ¾ or 1 in.

13 It includes three classes of work: *a*, chuck work, having diameter generally exceeding the length, and held wholly by a chuck or face plate; *b*, bar work, which is held in a chuck and steadied by back rests; and *c*, work having dimensions similar to bar work, but which must be turned on center points, with or without following and fixed steady rests.

14 It will be noticed that this excludes all of that larger and heavier lathe work in which the principal duty of the lathe is the rapid removal of the stock. In the particular branch of work under consideration the rapid removal of stock is important, but not paramount.

15 Although the field of work includes all kinds of steel and cast iron, this paper will deal only with the standard open hearth machinery steel of about 20 points carbon.

16 In work supported on centers and in chucking work, the connection between the work and tool includes a number of joints, both for sliding the tool in relation to the work, and for the rotation of the work. Each of these joints has more or less slackness, and each of the slides and other members is more or less frail in structure. With a mounting of this kind the cutting edge of the tool does not pass through the metal without swerving and flinching.

TYPE OF TOOLS USED

17 In the class of work under consideration each piece has several diameters, with shoulders which should be accurately spaced and formed. Nearly all the shoulders required in this class of lathe work are the so-called square shoulders.

18 In engine lathe practice these shoulders are "squared up" by a side tool after the other turning has been done by a round nose or diamond point tool, but in the turret lathe for bar work these shoulders are produced by the same tool that takes the stock removing cut.

19 The tool used in turners for bar work cuts on the same principle as the engine lathe side tool; that is, its rake or top slope is almost wholly side slope, and its cutting edge stands at an angle of 90 deg. to the axis of the work.

20 In the engine lathe a tool of this character has generally been unsatisfactory for rapid turning, yet in the turret lathe this very tool seems to be universally used for all bar work. The difference in performance seems to be due to the difference in mounting. It works well where there is no chance of vibration, but trouble begins

when it is used in a machine like the engine lathe or turret-chucking lathe, in which the work is supported by one part of the machine and the tool by another, and the true path of the cutting tool through the metal is dependent on the entire structure of the machine, there being nothing to prevent quivering.

21 The no-clearance tool to be described is a side tool without clearance. Its under face bears flatly against the work, thereby preventing the lateral quivering which has previously made this type of tool inefficient.

MEANS FOR IMPROVING EFFICIENCY

22 A machine's efficiency is proportional to its strength to resist its working stresses. There are two ways to increase this efficiency: *a*, by strengthening the machine; and *b*, by reducing the stresses for a given result.

23 In the writer's previous work the strengthening of the machine has been accomplished by the elimination of unnecessary features, and placing the necessary joints for obtaining the various motions in the least objectionable positions. But since this has been so fully outlined in a semi-commercial treatise entitled The Evolution of the Machine Shop, it is unnecessary to make further reference to the special forms of design therein set forth, except to say that a single-slide scheme of lathe design was adopted to eliminate the complicated and frail construction of the multi-slide tool carriage which is now in almost universal use in all standard machine tools.

24 The next step was to devise a means for minimizing the stresses at the cutting edge, and the object of the present paper is to explain how this result has been obtained.

25 This reduction of stresses may not be important in roughing work in which a flinching of the work or machine may be disregarded so long as the machine continues to crush off the metal, but for the kind of work mentioned in this paper it has been considered of first importance.

Cutting Stress

DIRECT CUTTING STRESS

26 For the purpose of analysis the cutting stress may be divided into three elements: the direct cutting stress, the separating stress, and the tendency to quiver, which we will consider in turn.

27 By direct cutting stress we mean that part of the stress that is directly downward in a lathe. With all other conditions unchanged,

we should expect to find that an acute-edged tool would offer the least resistance, and that the difference in direct cutting stresses for tools of varying cutting angles would show a marked reduction in favor of the more acute tools.

28 Dr. Nicolson's experiments below 60 deg., already mentioned, showed an increase in cutting stresses and a marked loss in endurance, but these tests were on dry cutting without the benefit of a lubricant or a cooling solution. The thin edge tool is undoubtedly benefited more than the blunt edge tool by lubricant or cutting medium. Just what cutting angle would be the best under conditions of most efficient cooling medium may not yet be fully known.

29 That there is no marked difference in cutting stress for the blunter tool of varying cutting angles really does not affect the situation when we try the real cutting or sliding angles, which may be roughly stated to be efficient in proportion to their acuteness.

30 It is obvious that the least direct cutting stress for a given depth and feed would be obtained by a straight-edge tool, and one that would take a chip in which there is the least molecular change.

31 Crushing and partially or wholly shearing the chip into chunks which are three or four times the thickness of the feed undoubtedly increase the working stresses and heat.

32 The cuts accompanying Dr. Nicolson's discussion, p. 333, vol. 28 of Transactions, clearly illustrate the great distortion that takes place even in cutting with an acute tool of 60 deg. and a straight edge. This tool does not have even the disturbing element of shearing action at the edge of the chip, but the experiment shows the distortion of nearly every part of the chip. A tool having a round nose or a blunt edge would doubtless show still greater distortion.

33 A flat top slope should have a straight cutting edge. The more the edge is rounded the greater the conflict of the metal crowding to the edge. The flow of metal on the top slope of the round nose does not move in one direction wholly, but tends to travel towards the center of the curve. The conflict of currents of metal which approach the center from various parts of the curved cutting edge increases the direct cutting stress.

34 The crushing process of the present scheme of turning is due both to the bluntness of the cutting angle and the shape of the edge. A curved edge should have a curved top slope in order to remove the chip with the least distortion of the metal. The curved top slope for this purpose would make the shape of the cutting edge similar to the cutting edge of a carpenter's round-nosed chisel. This form of

tool is not offered as a practical form, but is mentioned to emphasize the unnatural flow of the chip that must take place on the flat top slope of a round nose tool.

SEPARATING STRESS

35 By separating stress we mean that stress which, in turning a shaft, forces the tool outward radially. Increasing this stress causes the work and tool to move apart, and results in variation in diameter, also in an irregular and generally inaccurate product, particularly when the rough stock runs eccentric, or irregular. Although this separating stress may be lessened by giving the tool more back slope, this is possible only in tools taking light depth cuts. A lathe tool, however, which takes a cut like a side tool, gives little or no tendency to separate radially.

36 With the side tool set at an angle of 90 deg. to the travel of the feed, the feeding stress does not tend to force the work and tool apart; in fact, this tool may be set so as to produce a slightly beveled shoulder either side of the 90 deg., so as either to draw the work and tool together when making an overhanging shoulder or to force the work and tool apart when producing an external bevel.

QUIVERING STRESS

37 The quivering stress due to the nature of the chip is affected by the cutting angle of the tool. The chunks which make up the parts of a chip are less firmly united in a chip taken by a tool of 70 deg. cutting angle than by a tool of 50 deg., and of course the more firmly united chunks give a more continuous chip with the least vibration of stresses.

38 In turret lathe practice, especially in bar work, the tool and work are held together by a back rest which follows on the surface produced by the cutter, and in some kinds of turret-chucking work the tools for interior work are mounted on boring bars which take bearing either in the work or in the chuck which holds the work. When tools get this steadying support directly on or in the work, they are freed from the chattering due to the machine mounting, but not free from that due to their own frailty or to the intermittent flow of the chip as it is taken off in chunks.

RELATIVE DESTRUCTIVE EFFECTS OF HEAT AND LATERAL QUIVERING

39 The writer is not unmindful of the effect of heat in the destruction of the cutting edge, and fully realizes that no perfection of mounting of the work and tools will prevent destruction of the cutting edge of the tool by heat, but wishes to bring out the importance of the destructive effect of chattering which is ever present in standard types of machine tools. Heat is undoubtedly most destructive when roughing at high speeds, but the quivering plays a very important, if not the greatest part in edge destruction when finishing at the usual speeds.

40 Many machines are not run up to the high speed limit of the cutters. Even when provided with ample driving power, the strenuous life of attending a high speed machine is a little too much for the average man. As the speed is reduced, the quivering gains in relative importance, which should be taken into account in considering the no-clearance tool. With the slower speeds, tools should be used that give the best results at those speeds.

OTHER CONSIDERATIONS

41 The failure of the keen edge under normal cutting conditions, and its surprising endurance under some abnormal conditions, seem to indicate great possibilities open to any scheme that would maintain the best conditions. For instance, at one time, we have seen the edge of a diamond point broken off by an ordinarily heavy chip and at another time we have seen a similar tool deeply imbedded into the metal without breakage, the tool having taken a plunge and lifted or plowed up a chip of enormous proportions *without breaking the tool.* Every lathe hand has seen this performance. Usually it ends with breaking the tool or the center of the lathe, or both, but occasionally the lathe is stopped without breakage; then the lathe hand by great care may separate the work and tool without breaking the edge. The immense chip plowed up by a frail tool demonstrates what a cutting tool can do under some conditions.

42 We are also aware that under some conditions a cutting tool will actually sharpen itself in the process of cutting, yet neither of these results is regularly maintained. They suggest, however, the possibility of supplying a means by which they can be maintained in regular work.

CLEARANCE

43 Since the birth of the slide rest lathe, in which the tool was first guided by mechanism, turning tools have been given clearance and it has been assumed that they would not cut without clearance. Of course it is well known that the orthodox lathe tool goes out of commission after losing its clearance, but that does not demonstrate that a tool cannot cut without clearance. It only proves that the present tools require clearance as they are now formed and mounted.

44 A tool which has been ground for clearance, and set in such a position that its under face is at an angle to the shoulder produced, presents but a small area to the shoulder of work when the clearance of the extreme edge has given way. The area is so small, compared with the stress of the abrading metal passing it, that it rapidly scores and wears into a rough surface standing at a "negative" clearance angle. A tool with a negative clearance and rough surface quickly goes from bad to worse.

45 The tool which has by chance been set in an engine lathe so that a comparative large area of the under face rides on the wall of metal does not wear away, because its surface is not subjected to as great abrading pressure per unit of area. Its area is sufficient to withstand abrasion.

46 It was assumed by the writer that increasing the contact of the under face of the tool against the face of the work would make it possible to cut without clearance. The advantage of a no-clearance tool is that its face rides on a good area and supports the under edge against the pressure of the chip, thus relieving the edge from the one-sided pressure which must be borne by a tool having clearance. This one-sided pressure may be wholly or only partly relieved.

47 Of course, in all of the former types of tools the cutting edge must withstand the stress, which is wholly one-sided, excepting for the occasional condition stated, in which a cutting tool obtained by chance a bearing on its clearance face.

THE NO-CLEARANCE TOOL

48 In order to enable the tool to ride flatly against the wall of metal from which the chip is being removed, we have mounted it to allow a comparatively free swiveling action on a center line that is substantially coincident with the cutting edge of the tool. When the tool is so mounted the pressure of the chip on the top slope tends to throw the so-called clearance face against the shoulder, for the

mounting allows the tool to swing around to the angle that may be necessary to fit any work form, from a straight surface in planer work, and the nearly straight surface in work of large diameter, down to the angle of a helix obtained by the coarse feed on work of relatively small diameter.

49 A tool so mounted either swings automatically to adapt itself to angularity of feed, or may be swung by hand as soon as the cut is started. Its natural tendency holds it snugly against the metal, but the force may be varied from one that equalizes the stress on each side of the cutting edge down to a very slight stress which only holds the tool in no-clearance position. An important feature is that the tool is free to swing around to offset the unequal wear on the "clearance" face.

50 In the early experiments the cutters used were clamped rigidly in a holder, which in turn was pivotally mounted on a fixed holder. The cutting edge of the tool was so located as to stand exactly on the center line of the swiveling holder.

51 In the later experiments the scheme has been simplified by loosely mounting the cutter itself, providing it with a round bottom struck from a center line which is near the cutting corner of the tool. The cutting edge is usually standing at an angle to its center line of swivel, giving the tool a front slope. The scheme of inclining the cutting edge to the line of swivel was adopted for the purpose of using a bar-shaped tool in which its shape could be maintained by grinding, for with this shape grinding back the end provides for the wearing down of the top edge. This gives the tool a front slope when the swiveling center is kept horizontal. In some cases it may be well to tilt the holder to an angle that brings the cutting edge horizontal.

52 This departure from the ideal center position of the line of swivel is not sufficient to cause any trouble. In fact the pivotal line need not be exactly parallel to the cutting edge, neither is it necessary to have it very near the center line of swivel. It is probable that under some conditions the cutting edge may advantageously be located either above or below or on either side of the cutting edge. The exact location of the cutting edge relative to the center of oscillation partly determines the pressure with which the tool rides against the wall of metal from which the chip is taken.

53 The extreme top edge of the tool, in some instances, has been slightly flattened on the acuter angles, the flat measuring from about $\frac{1}{64}$ in. to $\frac{1}{32}$ in., and standing either 90 deg. from the so-called clearance face or sloping in either direction. Very good results were obtained

by giving it a negative side slope standing at a maximum angle of from 10 deg. to 15 deg. from the horizontal. This top flat seems to make a good resting place for the false edge, and it may be that its successful operation is dependent on the false edge.

54 One interesting phase of these experiments has been the comparative willingness on the part of the tool to relieve the carriage of the duty of feeding. This first became apparent when the carriage continued to advance after the feed had been "thrown out." This self-feeding feature, of course, cannot apply to the action of planers, boring mills, or work of large diameter. It is mentioned here only to indicate the absence of resistance to the feeding motion under some conditions.

55 The ultimate outcome of the use of acute angle tools may be to allow each tool to take a heavy cut on small diameters to determine its own feed. In the turret lathe this would be a distinct advantage.

CHIP LIFTER AND CHIP CONTROL

56 The chip produced by the acute angle tools is a continuous chip possessing great lateral strength. The continuous chip is preferred by any operator who has had experience with hot chips thrown off by tools of blunter angles, but while this particular feature enables him to observe the action of the tool closely without risk, the continuous chip in itself becomes troublesome, if allowed to run too long without breaking. In some of the first experiments with this tool, chips having a depth of about $\frac{3}{8}$ in., and produced by a feed of six to the inch, were found exceedingly troublesome, especially when allowed to run out to lengths of 5 to 15 ft.

57 The lateral stiffness of the chip of the more acute tool made it possible to increase the tearing open or splitting effect which occurs in cutting metals. To increase the tearing action it is necessary to allow the chip, after it has passed from the edge of the tool, to pass over a lifter in the form of a wedge, either formed integrally with the tool or placed in the path of the chip near the tool, having an angle that not only assists in tearing the metal ahead of the tool, but also relieves the slope of the tool near the edge from an important part of the labor.

58 In other words, a chip possessing lateral strength made it possible to carry an important part of the cutting or splitting action farther away from the extreme edge. The heat generated by this part of the work, because of its position, of course in no way reduces

the life of the extreme cutting edge. Experiments with the chip-lifting scheme seem to indicate that under ideal conditions the duty of the extreme edge of the tool may be simply to cut through metal which may be under more or less of a tearing or splitting stress.

59 Although this chip-lifting effect may be produced by a top slope having a curved surface, it has seemed best for the convenience of grinding the tool on an ordinary wheel to keep the top slope of the cutter a flat surface, and to introduce this chip-lifter as a separate member, either as a part of the tool holder or in conjunction with the chip-breaker to be described.

60 Although it is, as was stated, a satisfaction to be able to stand near the cutting tool and have some assurance of the direction in which the chip will travel, and to know that it is integral and not shooting out in hot chunks at all angles from the tool point, a continuous chip is nevertheless troublesome. Even with blunt tools, the curling chips which are sometimes used to illustrate ideal working conditions of a machine require the constant attention of the operator, and either a very large receptacle which doubles the floor space required for the machine or the almost constant attendance of an extra man for removing the chips from the room.

61 The use of the more acute angles increases the chip troubie, and may in some instances make it advisable to retain the blunt cutting angles, or at least, tools which produce tolerable chips.

62 For turning bar work in the turret lathe it has seemed best to adopt a chip-breaker which produces a fracture by placing an obstruction in the path of the chip at such an angle that the chip is bent, either by lifting or depressing, or both, shortly after it has left the tool,to an extent beyond its breaking point. In order to employ the chip-lifter most efficiently for the purpose of relieving the top slope of the cutting tool, the writer has preferred to use a chip-breaker which depended on depressing the chip after it passed over the chip-lifting incline. A breaker of this kind breaks the chip in lengths varying from ½ to 3 in.

CONCLUSIONS

63 The no-clearance cutter relieves the edge from the one-sided pressure.

It prolongs the life of the cutter by allowing abrasion on its face without producing negative clearance.

It prevents lateral quivering.

It converts the lip angle into cutting angle, which for a tool of given form constitutes a gain of from 5 to 10 deg. in cutting angle.

It has extended the working range of the side tool which gives the minimum separating stress.

It has made possible the use of acute-angled tools which reduce the cutting stress, thereby increasing the output of machines which have been limited by lack of pulling power.

The reduction of the cutting and separating stresses has increased the accuracy (or output, which is generally interconvertible with accuracy) on nearly all lathe work.

This reduction also increases the output, which has been limited mostly by the frailty or the slenderness of the work.

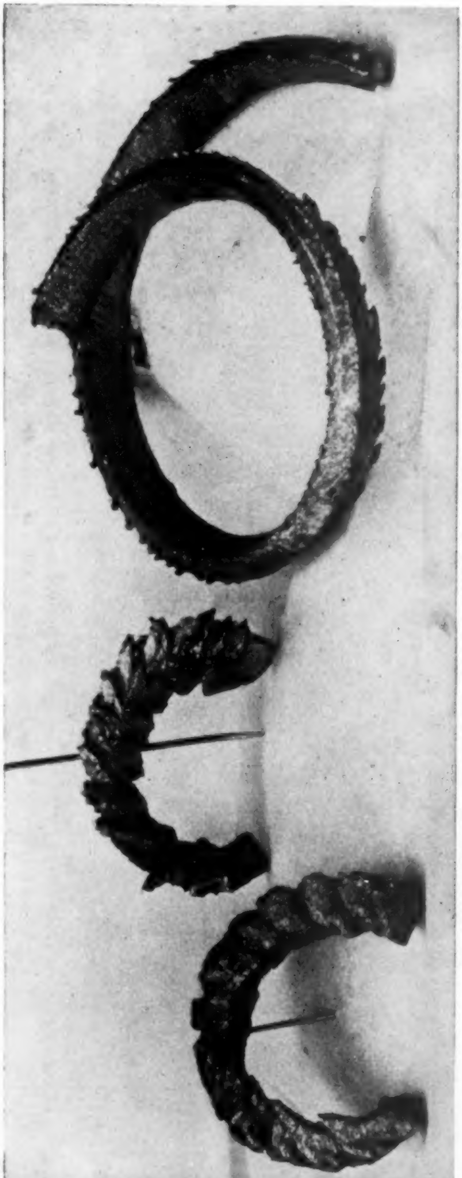

FIG. 1 CHARACTERISTIC CHIPS (ABOUT DOUBLE SIZE). CHIPS AT THE LEFT MADE BY DIAMOND POINT TOOL HAVING 70 DEG. CUTTING ANGLE. CHIPS AT THE RIGHT MADE BY NO-CLEARANCE TOOL, 45 DEG. CUTTING ANGLE.

Fig. 2 Samples of Chips and Cutters: Cutting Angles from Left to Right 45 deg., 60 deg. and 75 deg. These Chips were Produced in an Engine Lathe with Holder Shown in Fig. 12

THE CHIPS WERE CONFINED EDGEWISE BETWEEN THE BODY OF WORK AND THE END OF THE HOLDER. THE BREAKAGE OF CHIPS TAKEN BY THE 45 DEG. TOOL WAS DUE TO CHIPS GETTING CAUGHT BETWEEN THE WORK AND TOOL HOLDER, USUALLY DUE TO THE IRREGULAR WINDING OF THE CHIP. THE CHIPS PRODUCED BY THE 60 DEG. TOOL WOUND UNTIL THE CIRCLE WAS GREATER THAN THE CHIP COULD TAKE WITHOUT BREAKAGE.

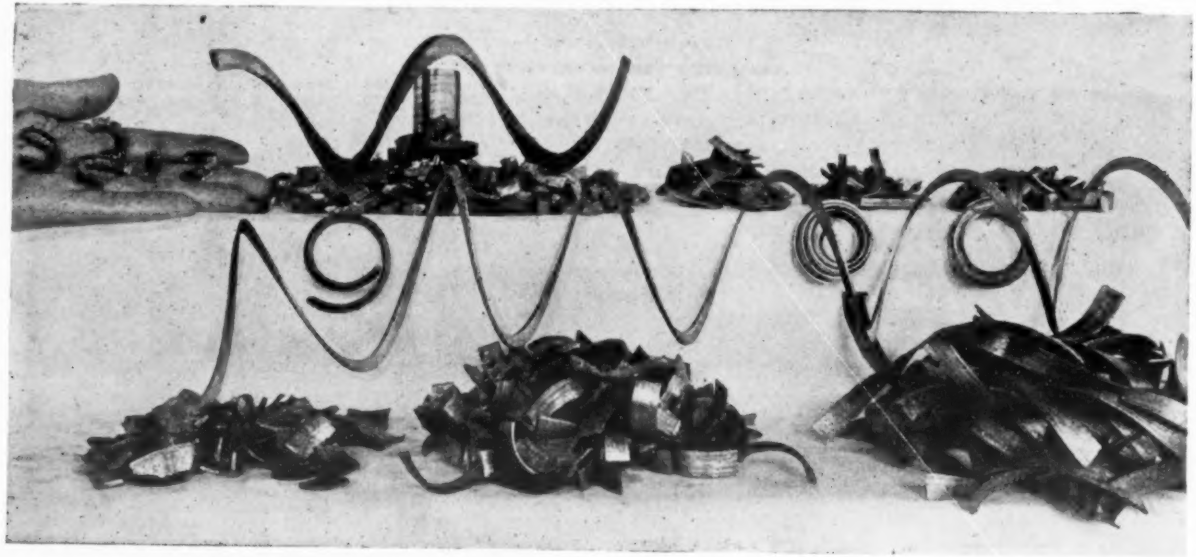

FIG. 3 SAMPLES OF CHIPS: THOSE HELD IN THE HAND WERE PRODUCED BY BLUNT SIDE TOOL HAVING 75 DEG. CUTTING ANGLE. ALL OTHER CHIPS WERE PRODUCED BY TOOLS HAVING CUTTING ANGLE OF 45 DEG. OR LESS.

THE CHIPS IN THE THREE SMALL PILES AT THE RIGHT ON THE TOP ROW AND THOSE ON THE LOWER ROW WERE BROKEN BY THE CHIP CONTROLLER SHOWN IN FIG. 15. THESE CHIPS WERE PRODUCED BY CUTTERS NO. 2 AND 3 IN FIG. 4 AND 5, RUNNING AT 40 to 45 FT. PER MIN. (PERIPHERY SPEED) AND 22 FEED PER INCH; DEPTH OF CUT $\frac{7}{16}$ IN., REDUCING FROM $1\frac{3}{4}$ IN. DOWN TO $\frac{7}{8}$ IN.

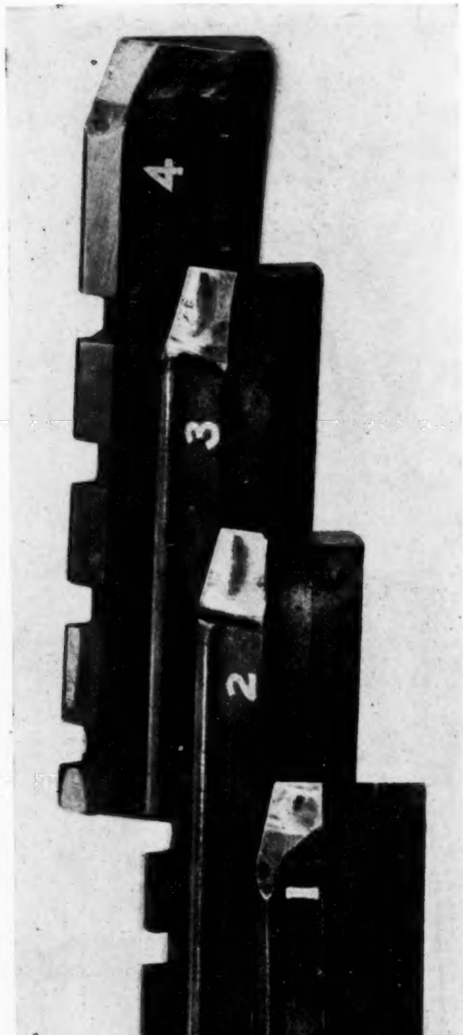

Fig. 4 Cutters Used in the Flat Turret Lathe

ILLUSTRATION SHOWS THE ABRASIVE CONTACT OF CHIP ON THE TOP SLOPE. NO. 1, 2 AND 3 WERE USED IN TURNER, FIG. 10. NO. 4 SHOWS ONE OF THE EARLIER FORMS.

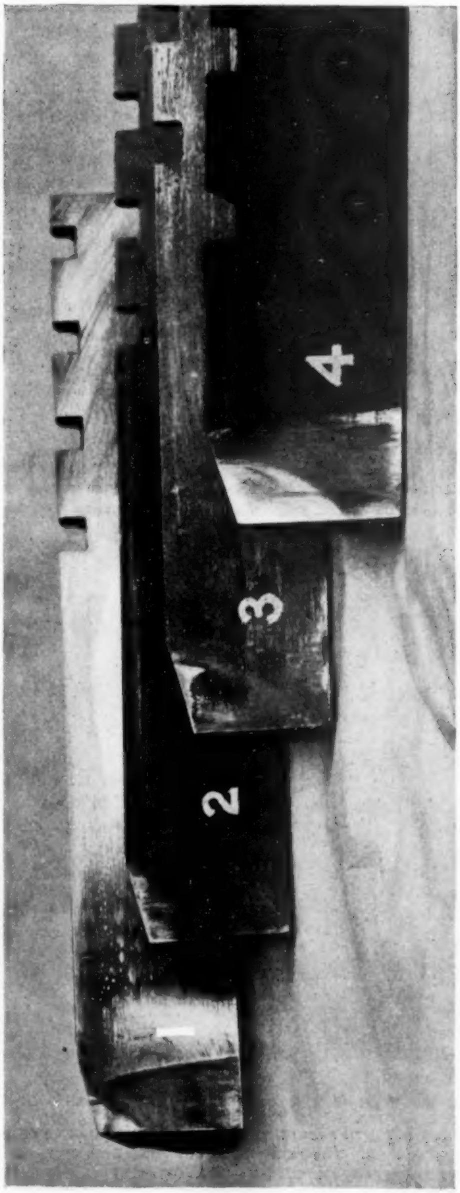

Fig. 5 Reverse Side of Cutters Shown in Fig. 4

Illustrating the rubbing contact of the tool against the shoulder of the work. Each tool bears the same number in both cuts.

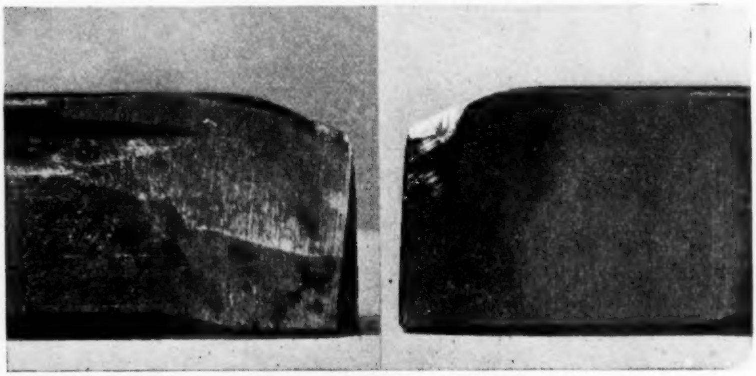

Fig. 6 No-Clearance Tool, Full Size

SHOWING AN EQUAL ABRASIVE EFFECT ON EACH SIDE OF EDGE. THE VIEW AT THE LEFT SHOWS THE TOP SLOPE, THE ANGLE OF WHICH WAS INCREASED BY CHIP ABRASION. THE VIEW AT THE RIGHT SHOWS THE ABRASIVE EFFECT OF THE SHOULDER OF THE WORK WHICH REDUCED THE CUTTING ANGLE, BUT NOT AS MUCH AS THE ABRASION OF THE CHIP INCREASED IT.

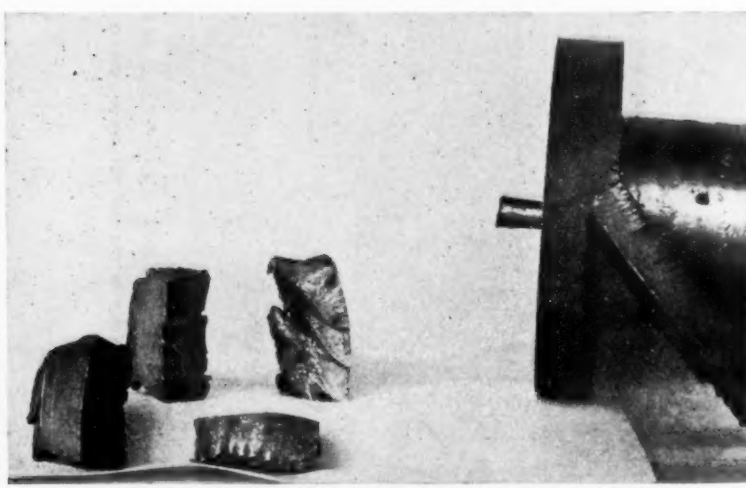

Fig. 7 Sample of Broken Chips and Work with an Unbroken Chip

THE VIEW IS ABOUT ONE-SEVENTH LARGER THAN SAMPLE, THE EXACT DIMENSIONS BEING 1¾ IN. DOWN TO ABOUT 1 IN. DIAMETER. THE FEED WAS ABOUT 7 PER INCH, CUTTING ANGLE OF TOOL ABOUT 38 DEG., EXTREME EDGE 1/32 IN. FLAT. THESE CHIPS WERE BROKEN BY A SCHEME SIMILAR TO THAT SHOWN IN FIG. 16.

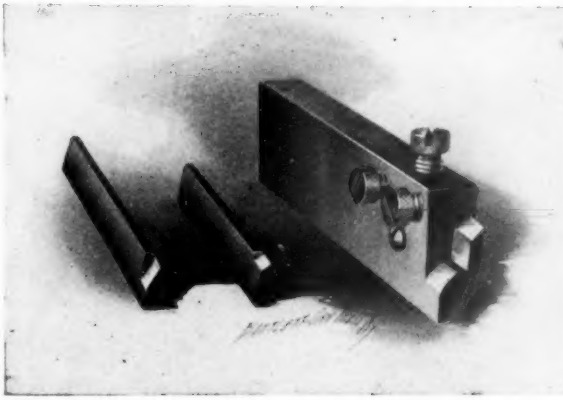

Fig. 8 No-Clearance Tool for Standard Engine Lathe Tool Post, with Three Cutters of Different Angles

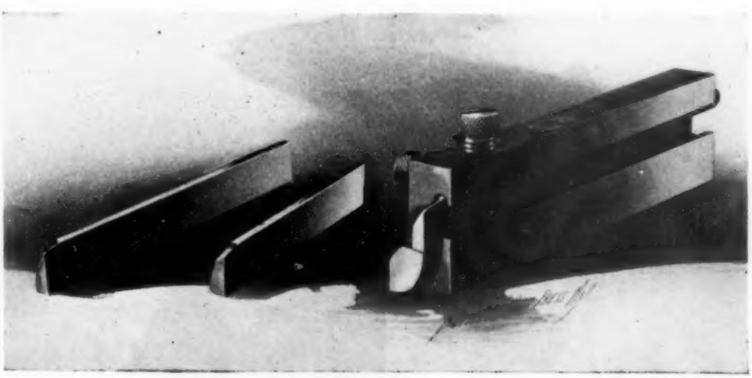

Fig. 9 View of Other Side of Tool Shown in Fig. 8

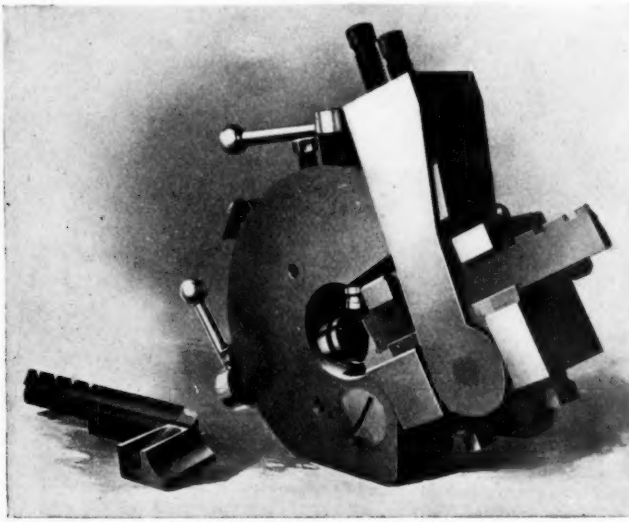

FIG. 10 NO-CLEARANCE TURNING TOOL FOR THE FLAT TURRET LATHE

THIS TOOL IS PROVIDED WITH A DOUBLE ADJUSTMENT BY WHICH IT MAY BE SET TO TURN TWO SIZES. SEE FIG. 13–15

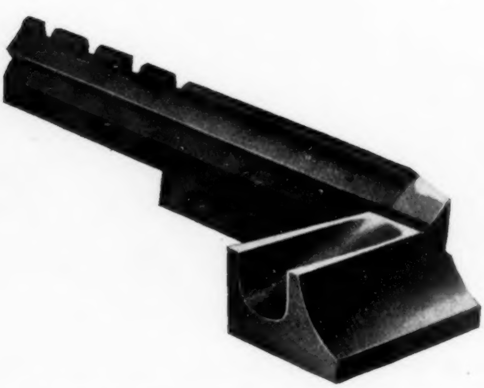

FIG. 11 LARGER VIEW OF THE CUTTER AND CHIP CONTROLLER SHOWN IN FIG. 10

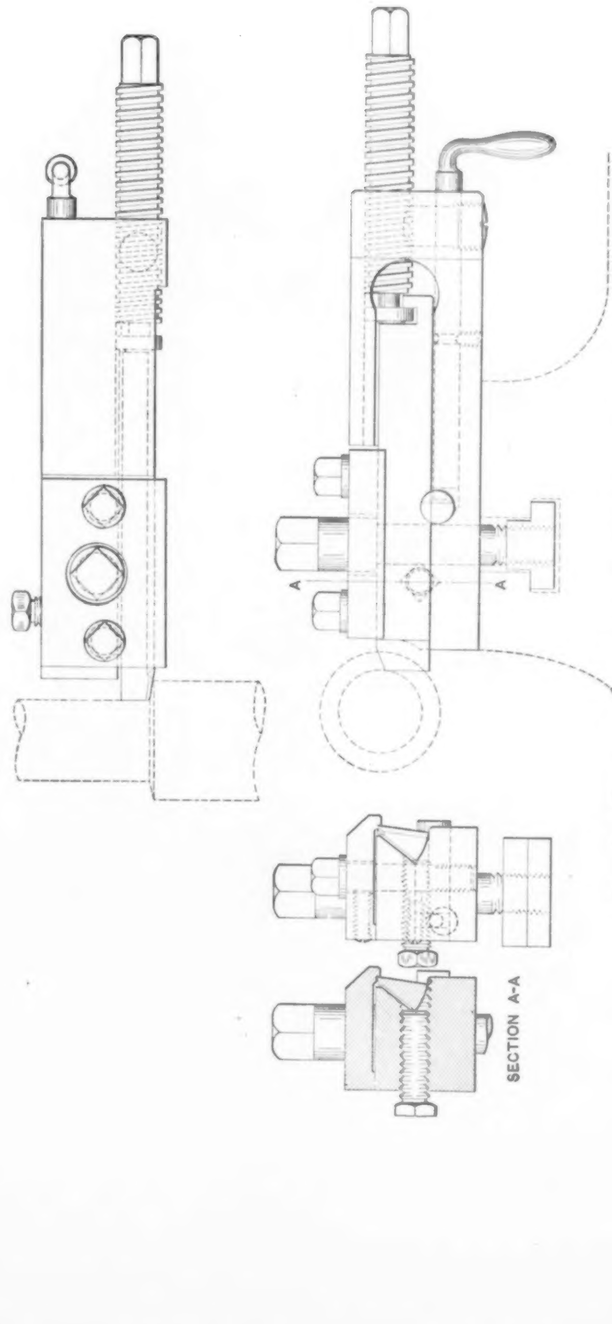

Fig. 12 Form of Turning Tool and Holder for Engine Lathe

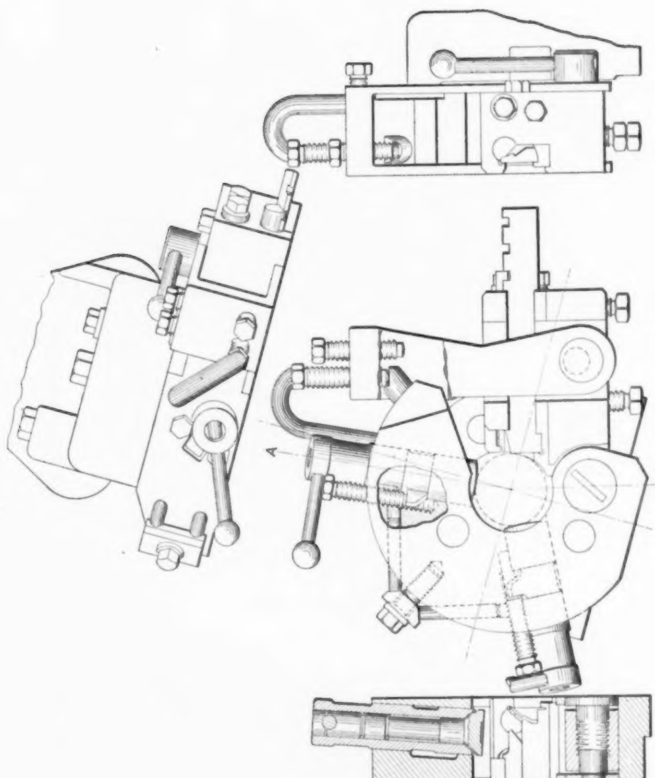

Fig. 13 Turning Tool for Flat Turret Lathes

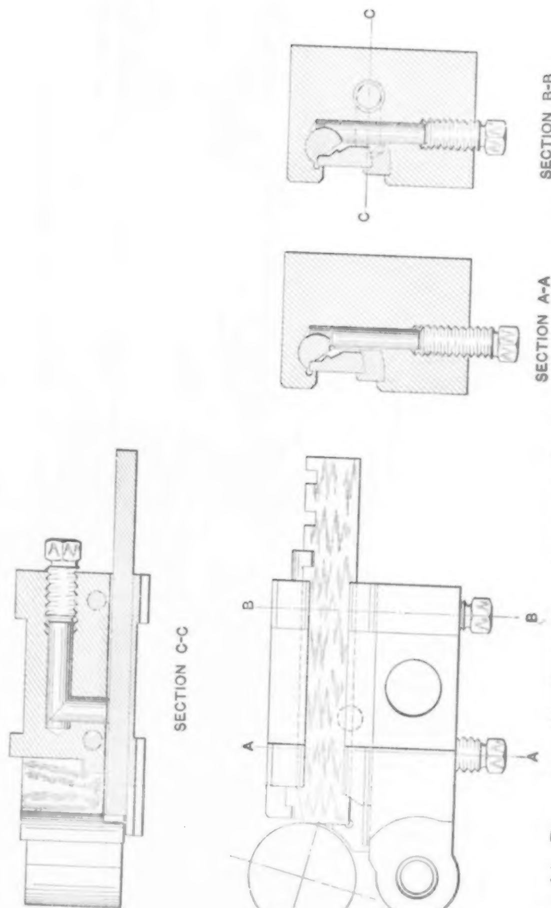

Fig. 14 Details of Turning Tool Showing Means of Holding Cutter and Limiting the Oscillation

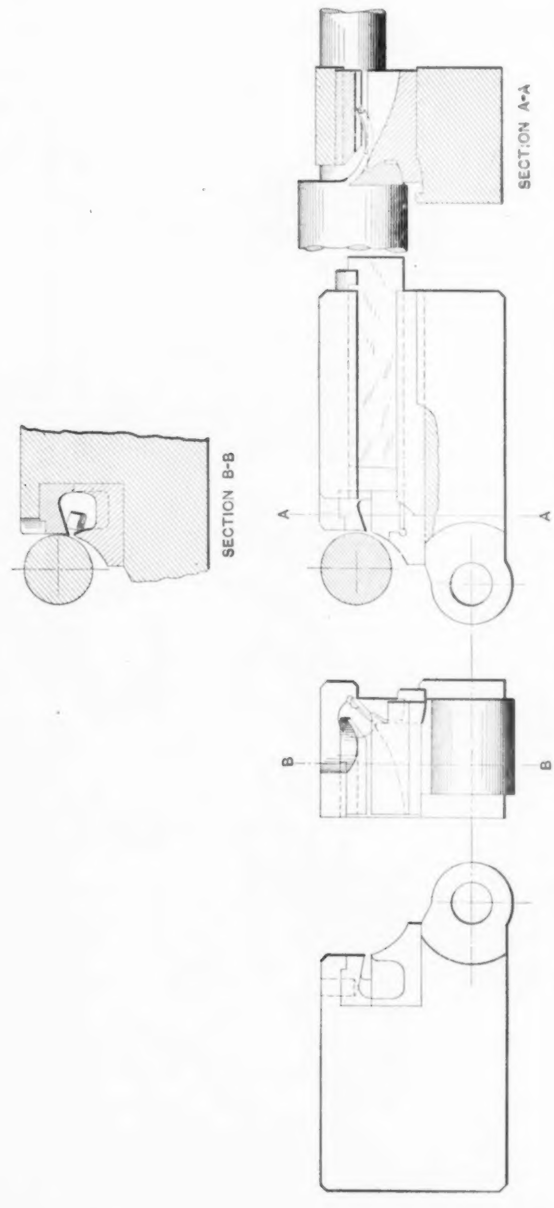

FIG. 15 DETAILS OF TURNING TOOL SHOWING CHIP BREAKER FOR A CURLING CHIP

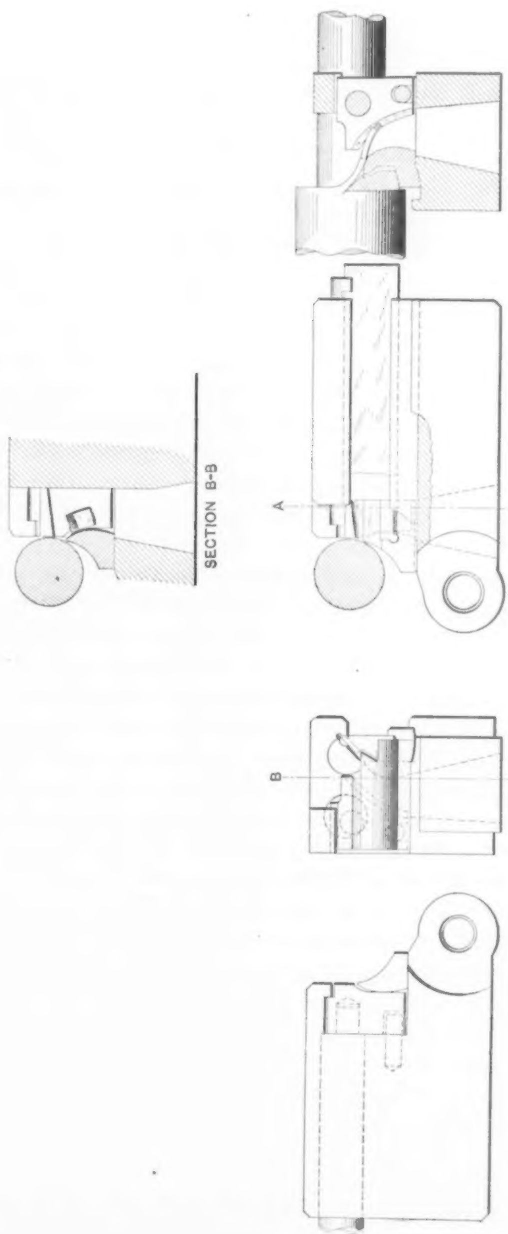

Fig. 16 Alternate Scheme for Chip Breaker for a Comparatively Straight Chip

DISCUSSION

Mr. H. H. Suplee This paper is entitled *Metal* Cutting Tools Without Clearance, and so my remarks may be a little off the exact subject. A number of years ago I used *wood* cutting tools without clearance in planing machines, for much the same purpose as Mr. Hartness has used his. In the wood planing machine, in order to avoid chattering and get smooth work on high grade hard wood flooring, and work of that kind, the knives were set as true as possible, and then the cutter head revolved, and a revolving emery wheel was carried back and forth, grinding off the back edge of the knife, to maintain the true cylindrical surface of the revolving cutter head. So a flat edge, or rather a cylindrical edge, was obtained for the back part of the cutting knife. The cutter produced remarkably smooth work. All the knives were doing an equal share of the work, and the chattering and quivering common in many planing machines was almost entirely removed. The real difficulty was that the edges of the tools got so hot after a certain period, that their temper was drawn.

The Author Mr. Suplee's remarks regarding the use of no-clearance blades in wood-planing machines are full of significance. The question of heat generated by the friction of the riding-contact is one that will require more time to settle definitely. The results may not be the same for different materials. We would expect an important difference between steel and cast-iron. The extra heat generated in wood-planing may have been due to some or all of the cutters having acquired a negative clearance. The springing of the material or the head would cause substantially a negative clearance. Excellent results are now being obtained by this scheme by the planers made by the S. A. Woods Company of Boston.

2 The whole subject is so full of unsettled points that the author hopes others will be induced to contribute papers, especially along the lines of acute cutting-angles, clearance, and cooling or lubricating solutions.

No. 1218

INTERCHANGEABLE INVOLUTE GEAR TOOTH SYSTEMS

By Ralph E. Flanders, New York
Non-Member

The standard form of gear tooth for cut gearing has a pressure angle of 14½ degrees and an addendum (in a one diametral pitch gear) of one inch or $1 \div P$. The pressure angle of an involute gear is the angle that the line of action makes with the common tangent to the pitch circles of two gears in mesh; or it may be stated as the angle made by the side of a rack tooth of a given system, with the perpendicular to the pitch line (see a in Fig. 4). In any system of involute gearing the pressure angle remains the same for all gears in the series. The addendum is the height of the tooth above the pitch line. As this also remains the same for all the gears in a series, a standard system of interchangeable involute gearing may be defined by giving the pressure angle and addendum.

2 While a pressure angle of 14½ degrees, and an addendum of one inch for a one diametral pitch tooth was formerly used almost universally for cut gearing, there have always been occasional departures from this standard. Of late years these departures have become more and more numerous. The pressure angles that have been used have run as high as 28 degrees in some cases, and the addendum has been made as small as 0.6 of the standard height. As this tendency to use shorter addendums or greater pressure angles appears to show no signs of abating, and has resulted in a considerable diversity of forms of gearing in different lines of work, it has seemed worth while to investigate the whole question, to see if these variations are justified; and if they are, whether their justification lies in peculiar conditions, or in a lack of fitness of the standard system for any large part of the work for which it has hitherto been used.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

3 As far as possible, the effect of varying the pressure angle and addendum on the various working qualities of gearing is here shown in diagrammatic form, so that the eye can readily grasp the effect of the variations in every case. The method of calculating the various diagrams is described in an Appendix,[1] so as to have the body of the paper as free as possible from mathematics, leaving nothing to consider except the results obtained. To any one disposed to question the methods followed in the mathematical investigation the appendix will give full information on this matter. For the sake of simplicity, in all the calculations the teeth are supposed to be of 1-diametral pitch. The pressure angles investigated range from $14\frac{1}{2}$ to $27\frac{1}{2}$ deg. Three heights of addendum are considered: 1.0, 0.8 and 0.6 in. respectively, for 1-diametral pitch. These limits are thought sufficient for the purposes of this paper. It should further be stated that in all calculations where it is necessary to assume the number of teeth in the smallest gear of a series, that number is taken to be 12, the same as in the standard system. The following reference letters are used:

S = height of addendum.
a = pressure angle.
N = number of teeth.
N_{12} = number of teeth in 12-tooth gear, etc.
n = number of teeth in continuous action.

EFFECT OF VARYING THE PRESSURE ANGLE AND ADDENDUM

4 The effect of varying the addendum and pressure angle is investigated with reference to the following practical considerations: interference; number of teeth in continuous action; side pressure on journals; strength; efficiency; durability; permanence of form; quietness and smoothness of action; suitability for practical cutting processes; and miscellaneous practical considerations.

5 Fig. 1 illustrates the effect of changing the addendum and the pressure angle on the question of interference. The lines rising toward the right in the diagram indicate the largest number of teeth in the gear in any given system of interchangeable gearing, which will mesh with a 12-tooth pinion without correction for interference with the flanks of the teeth of the latter. Thus, in a system of interchangeable gears in which $a = 17\frac{1}{2}$ deg. and $S = 0.6$, the dia-

[1] On file in the library of the Society.

gram shows that all gears having more than 50 teeth must be corrected to avoid this interference with the 12-tooth pinion. The lines rising toward the left indicate the minimum number of teeth possible in the smallest pinion in an interchangeable series, to avoid entirely the phenomenon of interference. Thus with the standard form in which $a = 14\frac{1}{2}$ deg. and $S = 1.0$, if 32 be taken as the number of teeth in the smallest gear of the series, instead of 12, it will not be necessary to correct the rack or any other member of the system for interference.

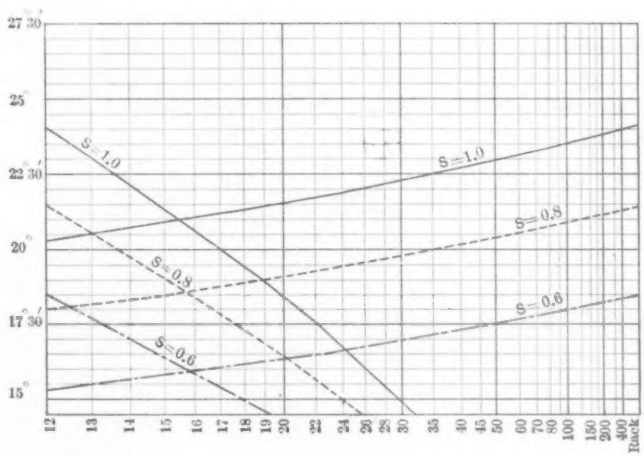

FIG. 1 EFFECT ON INTERFERENCE OF CHANGES IN THE ADDENDUM AND PRESSURE ANGLE

6 The following formulae were used in calculating this diagram. For the maximum number of teeth possible without interference with the 12-tooth pinion,

$$N = \frac{36 \sin^2 a - S^2}{S - 6 \sin^2 a} \tag{1}$$

For the minimum number of teeth possible without interference with an uncorrected rack,

$$N = \frac{2S}{\sin^2 a} \tag{2}$$

7 It will be seen from the diagram that with the standard form of gearing, in which a = 14½ degrees and S = 1.0, interference occurs to such an extent that the correction of the face of the tooth has to be carried clear down to the smallest gear in the series, it being impossible for two uncorrected 12-tooth pinions to mesh without interference. This condition is shown in Fig. 2. The contact between the two gears, running in the direction indicated and with gear A as driver, takes place along the line CD, being determined by the points of tangency of this "line of action," as it is called, to the "base of circles" of the two pinions. That part of the face of the teeth of gear A (see shaded portion at the right) which lies outside of the interference circle passing through point F, extends beyond any possible contact with the mating pinion, and so is useless for conjugate action in its uncorrected form. Not only is it useless but it is positively harmful as well.

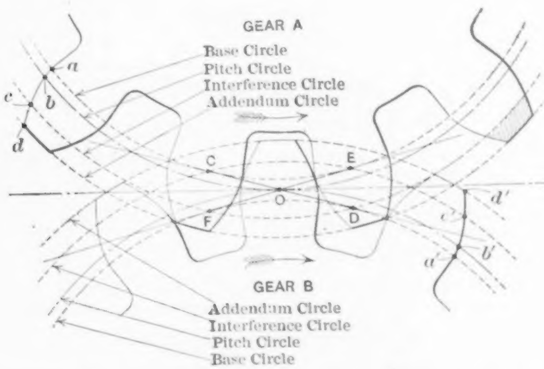

FIG. 2 EXTENT OF INTERFERENCE BETWEEN TWO 12-TOOTH STANDARD PINIONS

8 Fig. 3 shows, in diagrammatic form, the positions of the teeth of the two gears in Fig. 2 at the beginning and end of their action; contact begins at C and ends at D. As the gears continue to revolve, tooth face *aa* of gear A will interfere with tooth flank *bb* near the base circle, making proper meshing of the teeth impossible, no matter what the form given to the flanks of the teeth below the base circle. This phenomenon of interference is discussed in all treatises on gearing.

9 Everything outside the interference line then in Fig. 2 must be corrected for interference. The amount of this correction increases with the number of teeth in the gear, reaching its maximum in the rack, as shown in Fig. 4. As will be explained later, the nature of the

correction depends on the form arbitrarily given to the flanks of the teeth of the smallest pinion below the base circle.

10 The next thing to consider is the number of teeth in continuous action. This is at the minimum when the two gears having the smallest number of teeth allowed by the system are in mesh with each other and it is at its maximum in the case of the engagement of two racks.

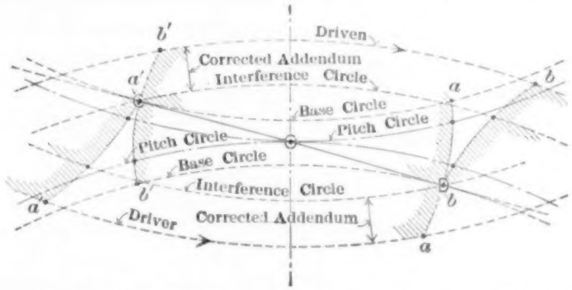

FIG. 3 LIMITS OF TRUE INVOLUTE ACTION WITH TWO 12-TOOTH STANDARD PINIONS IN MESH

A practical problem presented is that of finding the pressure angle and addendum that will give the required minimum action for the two smallest gears selected for the series. The diagram in Fig. 5 offers a practical solution of this problem. Suppose, for instance, it is required to find the minimum pressure angle that will allow 1.1

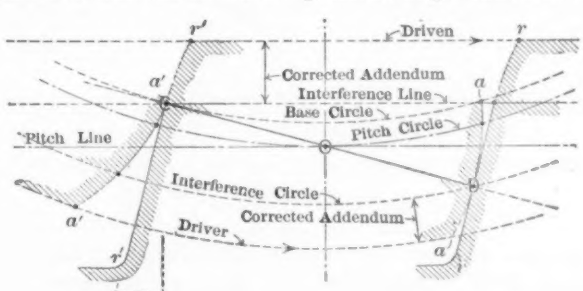

FIG. 4 LIMITS OF TRUE INVOLUTE ACTION WITH A 12-TOOTH PINION AND RACK (STANDARD) IN MESH

teeth to be in continuous action in a pair of 16-tooth gears, when the addendum is 0.8. (By saying that 1.1 teeth are in continuous action, we mean that the line of action is 1.1 times as long as is necessary for continuous action. At times, of course, two pairs of teeth will be engaged, while but one pair will be in action the remain-

der of the time.) The diagram shows that the pressure angle must not be more than 26½ deg. to make this possible.

11 It will be noticed that all the lines running toward the lower left-hand corner, for different values of S but the same values of n, merge with the line marked only for the value of n, the latter line extending downward toward the right. In reading a line for the given values of n and S it is to be traced downward to the left, to its junction with the full line for its value of n, which is to be traced downward. The point of junction of the two lines marks the point where interference ceases for a given value of n and S. For instance, if $n = 1.4$ and $S = 0.8$, the diagram shows no interference in the

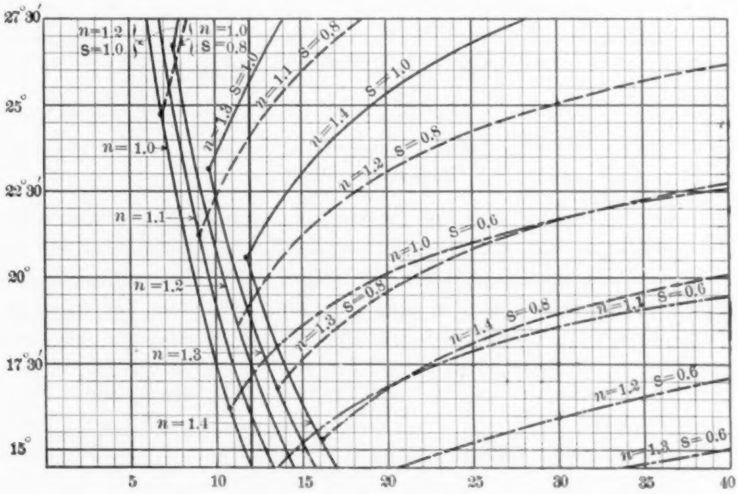

FIG. 5 NUMBER OF TEETH IN CONTINUOUS ACTION, IN MATING PINIONS OF THE SAME SIZE

case of the two smallest, or 16-tooth pinions meshing with each other, and so there is no interference in any gears of this series.

12 This change in the direction of the line indicates that many cases have two solutions. For instance, let it be required to find the proper pressure angle for 1.1 teeth in continuous contact, in a pair of 12-tooth gears, when $S = 0.8$. Following the curve $n = 1.1$, $S = 0.8$, we find that it crosses the 12-tooth ordinate at about 24 deg. 12 min. Following this curve down to its junction with the main curve, $n = 1.1$, we find that this crosses the 12-tooth ordinate again at about 16 deg. Thus there appears to be, as there are, two solutions to the

problem. These two results are shown diagrammatically in Fig. 6 and 7. In the first case, above the interference point, the action is limited by the points E and F, determined by the addendum circles for $S = 0.8$. In the second case, where interference occurs, the action is limited by points C and D, the points of tangency with the base circles. The amount of action with these two different angles is the same, there being 1.1 teeth in continuous contact in each case. In Fig. 6 the teeth are in conjugate action clear out to their points, while in Fig. 7 these points have to be corrected for interference. In Fig. 6, making $S = 1.0$ increases the number of teeth in contact by lengthening the line of action to E′F′. In Fig. 7, it will be seen, a similar increase of addendum makes no change in the number of teeth in contact, as the action is still limited by points C and D.

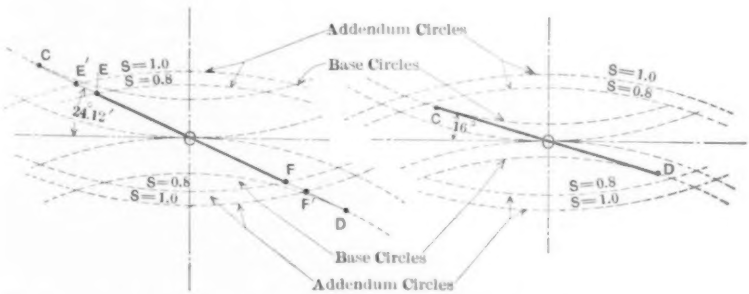

Fig. 6 Limitation of Conjugate Action by the Addendum Circles

Fig. 7 Limitation of Conjugate Action by Interference

13 The following formula was used for plotting the curves above the interference points in the diagram in Fig. 5:

$$N = \frac{(\frac{n}{2}\pi \cos a)^2 - S}{S - \frac{n}{2}\pi \sin a \times \cos a} \tag{3}$$

14 The following formula was used for plotting the curves below the interference point:

$$N = \pi n \cot a \tag{4}$$

15 Fig. 8 shows the maximum and minimum number of teeth in contact, for any system of interchangeable involute gearing in which the 12-tooth pinion is the smallest. The minimum contact occurs in the case of two 12-tooth pinions in mesh, and the maximum in the impractical case of two racks in mesh. As will be seen, the 14½-deg. stand-

ard series of whatever height of addendum, gives less than continuous action, being about 0.987. It will also be noticed that below the points of interference for the 12-tooth pinion (represented by the junction points of the maximum and minimum lines with the interference line) the contact is constant. Thus, for a system in which $\alpha = 20$ deg. and $S = 1.0$, the amount of contact between any two gears of the series from 12 teeth to a rack, is constant at about 1.40, while for a system in which $\alpha = 22$. deg. 30 min. and $S = 1.0$, the amount of contact varies between 1.36 for a minimum and 1.58 for a maximum. Fig. 3 and 4 show this condition, which indicates that if in any series there is interference in the case of the two pinions having the

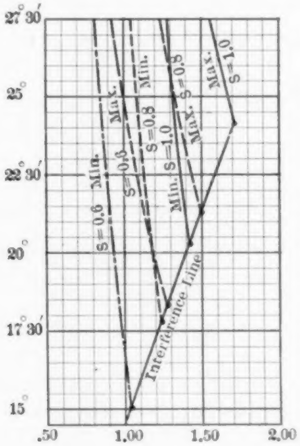

FIG. 8 MAXIMUM AND MINIMUM NUMBER OF TEETH IN CONTINUOUS ACTION

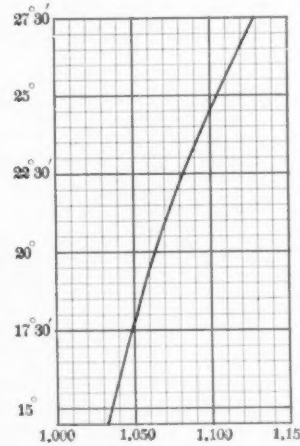

FIG. 9 SIDE PRESSURE ON BEARING

smallest number of teeth allowed by the series, the amount of action obtained in that case is constant for any other case throughout the whole series, up to that of two racks meshing with each other. As far as the author knows, this condition has never before been noticed.

16 In Fig. 3, which shows two minimum pinions of a series meshing with each other, the action is limited to the line CD. In Fig. 4, which shows the minimum pinion meshing with a rack, the action is still limited to the same line CD. It cannot extend beyond C in one direction, because it cannot pass the point of tangency in the base circle of the pinion. It cannot pass beyond D in the other direction, because the points of the pinion teeth are corrected beyond the

interference circle, losing their true involute form. In the case of any other gear meshing with a rack, the action is limited at C on one end owing to the correction for interference of the points of the rack teeth, and at D on the other end owing to the correction for interference of the points of the gear teeth. And in the case of any two gears, the action is similarly limited to the line CD by the corrections for interference at the points of the teeth. As may be seen, then, a true involute system in which $\alpha = 14\frac{1}{2}$ deg. and S = 1.0, just fails of continuous conjugate action in the case of any two gears of the series.

17 The formula used for calculating the maximum curves of Fig. 8, above the interference points, is as follows:

$$n = \frac{2\,S}{sin\alpha \times cos\ a \times \pi} \tag{5}$$

18 The formula used for calculating the minimum curves is as follows:

$$n = \frac{2}{\pi\ cos\ a}\left[\sqrt{(6 + S)^2 - (6\ cos\ a)^2} - 6\ sin\ a\right] \tag{6}$$

19 The formula for calculating the interference line, which gives the contact below the interference points, is the same as formula 4 when $N = 12$:

$$n = \frac{12\ tan\ a}{\pi}$$

20 One of the great practical advantages of the involute gear is the possibility of varying the center distance without interfering with true conjugate action. The comparative amount of separation possible for any two proposed systems, without losing continuous action, may be estimated from Fig. 8 by comparing the amounts of continuous action for the two cases. Naturally the one having the greater number of teeth in continuous action will stand more separation than one which has less. This point could be calculated, but by processes so devious that it did not seem worth while to spend the time for it.

21 With increase of the pressure angle there is an increase in the side pressure on the journals. Since the side pressure of the journals is the resultant of the tangential pressure between the mating teeth at the pitch line, and the radial outward thrust due to the angularity of the meshing surfaces at the pitch line, it varies directly with the secant of the angle. The curve shown in Fig. 9 is therefore a secant curve.

22 The rational formula for the strength of gearing is the well known one developed by Mr. Lewis.[1] For diametral pitch measurements, this formula takes the following form:

$$W = \frac{SFy}{P}$$

in which W = the load transmitted by the teeth in pounds.
S = the safe working stress of the material.
P = the diametral pitch.
F = the face in inches.
y = a factor depending on the form of the teeth.

As the strength varies directly with y, this factor forms the basis of comparison for different forms of teeth.

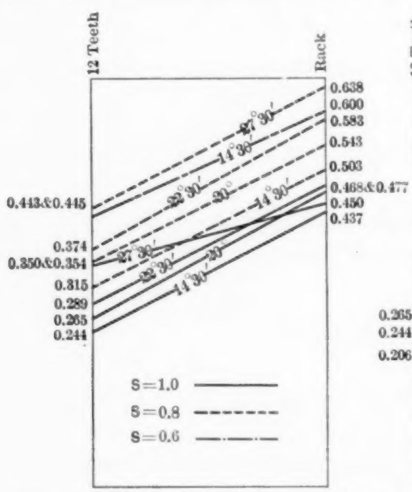

FIG. 10 MAXIMUM AND MINIMUM VALUES OF STRENGTH FACTOR FOR TYPICAL CASES

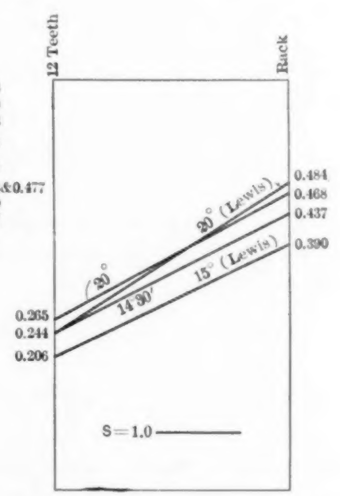

FIG. 11 COMPARISON OF VALUES FOR y HERE OBTAINED, WITH THOSE CALCULATED BY MR. LEWIS

23 Fig. 10 shows the maximum and minimum values of y for certain selected forms of gear teeth. It will be seen that the values of y rise rapidly with the decrease of the addendum, and more slowly with an increase of the pressure angle, which does not affect it so much.

24 This y factor was obtained by the method laid down by Mr. Lewis, with the incorporation of a slight refinement, taking into account the fact that the pressure is applied to the teeth at a point

[1] Kent's Pocket Book, p. 901.

above the pitch line. In order to get the values of y in the case of gears in which there is interference, the form of the flank of the 12-tooth pinion below the base line had to be arbitrarily determined. The proper corrections of the faces of the teeth were obtained by considering the flanks of the 12-tooth pinion to be radial below the base line. This gave to the rack teeth a cycloidal form above the interference line, generated by a rolling circle of half the pitch diameter of the 12-tooth pinion. In order to get the proper form of the fillet on the 12-tooth pinion, the rack tooth was lengthened by an amount equal to the clearance, and the corner of this extended tooth was rounded with a radius equal to $\frac{2}{3}$ of the clearance. The pitch line of this extended tooth was rolled on the pitch line of the 12-tooth pinion, and the fillet thus generated used in obtaining the factor y. As the shape of the fillet is of great importance in strengthening a gear tooth, it was thought best to determine it rationally as just described, its form being that which would be given it by a generating process such as that of the hobbing machine.

25 Fig. 11 shows that the values obtained for y vary somewhat from those obtained by Mr. Lewis (it will be noted that his values have been multiplied by π to make the formula read for diametral pitch). This variation is probably due to the fact that in this investigation the outlines from which the factors are calculated have their teeth corrected for interference, thus bringing the normal pressure at a greater angle at the points of the teeth; and to the fact that the fillets have been constructed by the method just described.

26 The work lost in friction in the case of two gears meshing with each other is proportionate to the product of the rate of sliding and the normal pressure on the surfaces in contact. This rate of sliding varies directly with the distance of the point of contact from the pitch point 0 (Fig. 3 and 4, etc.) of the gears. The pressure is constant throughout the action, except that when two teeth are in contact it may be considered that the pressure is evenly divided between them. The diagram in Fig. 12 was calculated by a modification of the method devised by an English engineer, Mr. Bruce,[1] the modification being simply that of considering the pressure on two sets of teeth as evenly divided between them, when there are two sets of teeth in continuous action. His method also needed correction to allow for the increase of pressure with the increase of α, and to take into account the time scale for distances on the line of action. The

[1] American Machinist, October 10, 1901

calculations are made for a single case of gearing, an 18-tooth pinion driving a 60-tooth gear. This case was selected as typical.

27 The diagram for durability is shown in Fig. 13. The wear between the teeth of two gears is proportionate to the continued product of the pressure, the rate of sliding, and a third factor which depends on the shape of the surfaces in contact. This factor is greater for sharply curved convex surfaces and less for a large radius curve rubbing on a flat surface, for instance. This being the case, it seemed to the author that a reasonable basis of comparison for different systems of gearing could be made by multiplying the values given

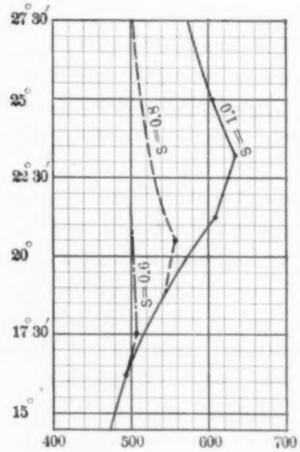

Fig. 12 Comparative Amount of Lost Work

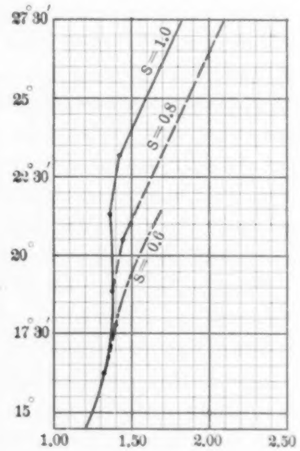

Fig. 13 Comparative Durability

in the diagram of Fig. 12 by a factor determined from the shapes of the surfaces in contact, and plotting the reciprocal. In determining this factor, the author has followed Mr. C. H. Logue,[1] who obtains it by adding together the sums of the curvatures (that is, the sums of the reciprocals of the instantaneous radii) of the gear teeth at the pitch line. This should give a value which is approximately an average of the conditions existing throughout the entire action. (It should be understood that Fig. 12 and 13 indicate comparative values only. Positive calculations of efficiency or durability are not here contemplated.)

[1] American Machinist, February, 1908.

28 The question of permanency of form may be considered aside from that of durability. By durability we mean the lasting quality of the gear, without reference to whether or not it keeps its shape in the wearing process. A gear which has permanency to a high degree might wear out very rapidly, but it would retain nearly its true form throughout the whole of its short life. In Fig. 14 are given a series of diagrams showing the tendency of a correctly formed tooth to wear at different points of the tooth outlines, in the case of a 12-tooth pinion and the rack, and for the same pressure angles and addendums that were selected for the strength diagram of Fig. 10.

29 Besides the various factors for which we have given diagrams, there are a number of important ones of such a nature as to be incalculable. One of these factors is the suitability of a system for use in different methods of tooth cutting. Any form of tooth which involves interference is very unsatisfactory for use in any generating process for instance. The formed cutter process works with equal facility on any form of tooth, except that the increase in the pressure angle gives somewhat more side clearance, leading to a freer cutting action.

30 The vital incalculable factor is smoothness of running. Smoothness of action depends theoretically on perfect conjugate action between the mating teeth. In practice it depends as well on accuracy of cutting tools and accuracy of machine setting. The cutter may be set out of center, or set deeper than required, or have its face ground at an angle considerably away from the radial plane required to give the true cutting shape. The standard form of gear tooth, developed by the Brown & Sharpe Manufacturing Company, has been improved by long practical experience to a point where it takes care of these practical inaccuracies in a very satisfactory way. This is possible, it appears to the writer, on account of the considerable portion of the outline which is indeterminate in form, namely, that part of the outline outside of the interference circle and inside of the base circle (see Fig. 2). These can be given any shape shown by practice to be proper for leading the teeth gently into action with each other at high speeds even with inaccurate cutting. Of course this shaping is entirely founded on experience.

31 It might be mentioned that the inaccuracies just mentioned as occurring in practical gear cutting are those chiefly met with in the formed cutter method. The inaccuracies of the generating processes are mostly due to difficulties in forming the cutters, and in taking care of the strains in the machine without deflecting the mechanism.

Looked at from this standpoint, it may be that the standard form is best for formed cutter work, while larger angles and shorter addendums are more suitable for generating processes.

RELATIVE IMPORTANCE OF THE VARIOUS CONSIDERATIONS

32 As to the importance of the various considerations mentioned above as being affected by changes of the pressure angle and addendum, it may be said that the difficulty due to interference (apart from its effect on strength, efficiency, etc., which is considered under those heads) is that of making the shape of the gear indeterminate. The design of the standard gear is empirical. While the writer has followed the plan of making the flanks of the 12-tooth pinion radial, and generating the fillet by means of an extended and rounded rack tooth, produced by such a radial flank, he does not know that this is the form of the standard tooth. The exact form is known only to the makers of standard cutters and is not public knowledge. It will be seen that this matter of interference makes of the standard system practically a short tooth system so far as the theoretical bearing is concerned. This is plainly shown in Fig. 14, for Case 1, which is the standard involute gear tooth. The bearing extends over but a very small part of the tooth face and flanks. The bearing can, of course, be carried clear to the points of the teeth by making the non-involute parts of the teeth on some other conjugate system, as was done, for instance, by the author in par. 24 by the cycloidal corrections. This would give more action than Fig. 8, but since only a part of it is involute, the excess of action would be lost as soon as the center distance is changed. A conjugate correction of this kind is made by the manufacturers of standard cutters.

33 The number of teeth in continuous action, shown in the diagram of Fig. 8, is of importance principally in relation to smoothness of action. It is not generally considered possible to count on distributing the load evenly between two teeth in calculating the strength of gearing, though it will be noted that where n is slightly more than unity, two teeth are in bearing at the beginning and end of the action, when the fiber stress in the teeth is at the greatest. In the matter of smoothness of action, it is necessary to have one pair of teeth take up the load before the previous pair drop it. Increased smoothness of action, due to smoothing out irregularities in cutting, could be effected by having two, three or more sets of teeth in action at one time, but this is impracticable with interchangeable involute gearing. A

considerable length of contact is also of advantage in permitting a considerable variation in the center distance, without loss of continuity of action, as previously explained. In this respect the increased pressure angles will be seen to have considerable advantage over the standard form.

34 The side pressure, shown in Fig. 9, is an almost negligible factor. As may be seen, the increase for even so great an angle as 27½ deg. is only 9 per cent above that given by the standard form. At

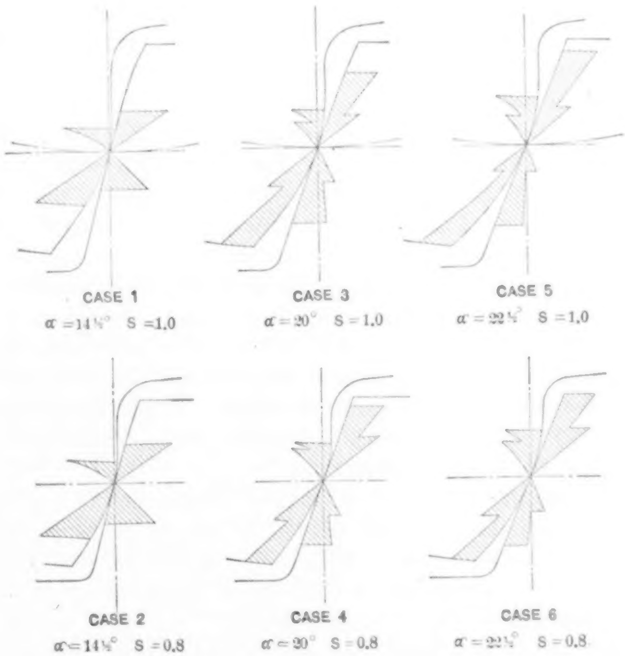

Fig. 14 Permanency of Form

22½ deg. it is 3⅓ per cent above. As the shaft bearings can take care of this slight increase of side pressure in a perfectly normal and satisfactory way, the question scarcely enters into consideration.

35 The matter of strength is an important one. If a stronger form of tooth can be used, many mechanisms can be gotten into a much smaller space than is otherwise possible. A case in point is the design of geared speed and feed mechanism for machine tools, in which the space problem is usually serious. By decreasing the

face of the gear to correspond with an increase in the strength factor y, the problem of the designer would be greatly simplified. Instead of reducing the width of the gear by using a stronger form of tooth, advantage can be taken of the stronger form to increase the number of teeth and make them of finer pitch. This gives smoother action, owing to the small inaccuracies in the fine teeth. Smoother action at high speed also means increased strength, owing to the lessening of stresses due to impact. It is becoming recognized that impact at high speeds is greatly affected by the accuracy of the surfaces in contact; and even where the strength is sufficient the matter of noise in high speed gearing is of great commercial importance. People would not buy noisy gearing if they could help it, even if it fulfilled their requirements in every other particular.

36 The question of lost work (plotted in the diagram of Fig. 12) is not a matter of serious moment. The efficiency of well-made spur gearing is so high[1] that the slight variations indicated by the diagram are of small practical importance. This diagram of course takes no account of the increase in lost work due to increase in journal pressure, which is an inconsiderable factor with well made bearings.

37 Durability is of much more importance. Gears have been almost invariably designed hitherto for strength rather than durability, but it is becoming recognized that in many cases the principal factor in gear design is the wear of the teeth. If then the methods by which the table of Fig. 13 is calculated are rational, the results there shown are of considerable importance.

38 Permanency of form is another important factor. Gears which the writer has seen worn out in hard service, show very plainly the severe concentrated wear inside the pitch line indicated in Case I of Fig. 14. The advantageous effect of an increased pressure angle on permanency is worthy of consideration.

39 The fitness of a tooth form for various cutting processes is of importance in determining the future of the apparently attractive generating methods of cutting gear teeth. One form of generating process has come into extensive use in spite of the handicap of the $14\frac{1}{2}$-deg. angle, with the various corrections for interference required. There is reason to believe, however, that this process works better with forms of teeth in which the angle is increased or the addendum decreased. In the matter of the hobbing process, which has developed so rapidly in the past two or three years, the author must say that

[1] Mr. Lewis' experiments, Kent's Pocket Book, p. 899.

he has seen comparatively little gearing cut by this process that seemed entirely satisfactory. This opinion will doubtless be disputed by all the makers and most of the users of these machines; but the difficulties of the process are recognized by everyone concerned, and many of them would be removed by a change in the form of tooth. Were the change advisable for other reasons, this fitness for generation would be an added advantage, though it would scarcely be worth changing on this score alone. So far as cutting teeth by the formed cutter process is concerned, an increase in the pressure angle is favorable, as more side clearance is obtained for the cutter; while decreasing the addendum gives a less volume to cut out, and a consequently cheaper production of gears. It might also be mentioned in this connection that the decrease in face or pitch that could be obtained by the use of stronger forms of teeth would also result in cheapening the gear, owing to the fact that less metal would have to be removed.

TYPICAL DEPARTURES FROM THE STANDARD FORMS

40 Among the various departures from standard practice which have been made from time to time, and especially in recent years, the writer will mention a few typical cases. The first is that of William Sellers & Company who have for many years used a form of interchangeable gearing in which $S = 0.942$ and $\alpha = 20$ deg. This form was determined by Mr. Wilfred Lewis, and is said to have given continuous satisfaction. A form of gear in use in the practice of the C. W. Hunt Co. retains 14½-deg. pressure angle, but uses a shorter addendum in which $S = 0.785$. While the Sellers standard was adopted principally for the avoiding of interference and for the increase of strength obtained, the Hunt standard is based on the increase of strength alone, which is, as shown by the diagram in Fig. 10, more effectively obtained by reducing the addendum than by increasing the pressure angle. Mr. Hunt states that this gearing has given unexceptional results in smoothness of action, as well as being stronger. What is practically the Sellers form of tooth, except that $S = 1.0$, has recently come into limited use in machine tool design, particularly in the case of the geared drives for heavy duty spindles.

41 In rolling mill work it has been the practice for many years to use pressure angles as high as 28 deg., which angle has been employed, the author understands, by the Carnegie Steel Company. 22½ deg. is a very common pressure angle. The addendum varies, being

sometimes of standard height and sometimes considerably less. A typical example is that of the Wellman-Seaver-Morgan Engineering Company of Cleveland, O. Through the courtesy of Mr. Stratton, Engineer of Construction, the writer has received blue-prints of the lay-out for the gears of a 22-in. blooming mill, in which the pressure angle is 20 deg. and the addendum 0.785. Even shorter forms of teeth have been used by this company in their gearing for their automatic water hoist, in which the pressure angle is 20 deg. and the addendum about 0.6. In rolling mill work in general, of course the prime consideration is strength, and this is undeniably obtained by such departures from standard form.

42 A form of tooth that has found wide use, particularly in automobile work, is the so-called "stub tooth," developed by the Fellows Gear Shaper Company. In this the pressure angle is 20 deg., while the addendum varies from 0.7 to 0.8, depending upon the pitch of the gear. While the particular way in which this variation of the addendum is taken care of is very convenient for the operator of this builder's machine, it seems to the writer a mistake not to have made it constant for all pitches. This is aside from the question of its general suitability however.

43 In reply to letters sent out to automobile builders (all the members of the American League of Automobile Manufacturers), the writer received thirteen replies, in which seven stated that they were using the "stub tooth" while four were using the standard form. Of those using the special gearing five stated that they had proved in service its ability to run as smoothly as the standard form, and of these, two had obtained even better results from it in this respect. Smoothness of action was not mentioned in the inquiry, so that the information on this point was purely gratuitous.

COMPARISION OF TYPICAL INVOLUTE TOOTH SYSTEMS

44 It is proposed to give, in connection with this discussion of the effect of the variation of pressure angle and addendum, a comparison, by the diagrams herewith presented, of various typical forms of gearing in relation to the practical points just discussed. For the purposes of this discussion the following forms are considered:

Case 1: $\alpha = 14\frac{1}{2}$ deg., $S = 1.0$.
Case 2: $\alpha = 14\frac{1}{2}$ deg., $S = 0.8$.
Case 3: $\alpha = 20$ deg., $S = 1.0$.
Case 4: $\alpha = 20$ deg., $S = 0.8$.
Case 5: $\alpha = 22\frac{1}{2}$ deg., $S = 1.0$.
Case 6: $\alpha = 22\frac{1}{2}$ deg., $S = 0.8$.

45 Case 1 has the standard dimensions. Case 2 is practically the Hunt system; 3 is the Sellers system; and 4 is practically the "stub tooth" system. The last two columns show the effect on the Sellers and "stub tooth" systems respectively, of increasing the pressure angle from 20 to 22½ degrees. The table, in which these various forms with their corresponding advantages and defects are tabulated, suggests that by a change in the present standard of gearing, marked advantages could be obtained on the following scores: avoidance of interference, giving a greater number of teeth in true involute action, and

COMPARISON OF SELECTED EXAMPLES OF INVOLUTE GEAR TOOTH SYSTEMS

Point of comparison	Case 1 Standard addendum and pressure angle	Case 2 Hunt standard (approximate)	Case 3 Sellers standard (approximate)	Case 4 "Stub tooth" (approximate)	Case 5	Case 6
Smallest pinion in series, to avoid interference	32	26	17	14	14	11
Maximum number of teeth without correction for interference with 12-tooth gear	None	None	None	36	35	Rack
Maximum and minimum number of teeth in continuous contact	0.98	0.98	1.38	1.38 1.18	1.58 1.36	1.44 1.13
Proportion of side pressure on bearing to tangential pressure	1.033	1.033	1.064	1.064	1.082	1.082
Strength factor of rack	0.437	0.503	0.468	0.543	0.477	0.583
Strength factor of 12-tooth gear	0.244	0.315	0.265	0.354	0.289	0.374
Comparative loss of work from friction	475	475	572	552	628	532
Comparative durability	1.21	1.21	1.39	1.44	1.40	1.63
Permanency	See Fig.	14				

For incalulable factors, see text.

permitting more separation with continuous contact; increase of strength; increase of durability; increase in permanency of shape; and increase in fitness for generation and cutting by formed cutters. On the other hand, such changes means an increase in side pressure, comparatively little change in efficiency, and an incalculable and unknown change in smoothness of action. Side pressure and efficiency (as has been shown) are of comparatively minor importance. Smoothness of action is of prime importance. The results of a change,

in this respect can only be arrived at by learning the experience of the users of various forms of gearing, and the author hopes for a full discussion in order that every known fact relating to smoothness of action may be brought to bear on this question.

46 Whatever the condition relative to high speed work, however, it seems to the author that the preceding discussion points clearly to the wisdom of an alternative gear tooth standard of shorter addendum and increased pressure angle for such work as heavy mill gearing, and slow speed gearing in general in which smoothness of action is not a prime requirement. Increased strength and permanency of form would appeal particularly to engineers whose work permits them to design machinery by rational rather than by empirical methods.

47 Aside from the specific cases mentioned, correspondence and conversation with makers of cut gearing bears testimony to a growing demand for, and use of, a stronger form of tooth. The adoption of a standard for this work would give all the advantages which come from standardizing: the preparation of tables of dimensions and of strength; the devising of odontographic tables for the fillet at the root of the tooth, and for the correction—if any is needed; and the reduction in the stock of form tools necessary for doing a general line of gear cutting work.

48 In any event, a standard of gearing of known form, which can be laid out by any engineer who desires to employ it, would be an advantage, if such a consummation is possible without sacrificing any of the good qualities which we are accustomed to expect from the present standard.

DISCUSSION

Mr. Wilfred Lewis I think Mr. Flanders has presented a very illuminating and complete analysis of the conditions governing the intelligent use of interchangeable involute gearing from a 12-toothed pinion to a rack. It appears from this that the so-called standard $14\frac{1}{2}$ deg. system is really no system at all, but rather an evolution by a sort of trimming process from the errors of the past.

2 About thirty years ago when I first began to study the subject, the only system of gearing that stood in much favor with machine tool builders was the cycloidal in which the describing circle was made one-half the diameter of a 12-toothed pinion, thus giving it radial flanks. At that time I was called upon to investigate a cutter forming machine in which the faces and flanks of the cutter were approximated

by circular arcs. I soon came to the conclusion that the true form of such cutters could not be approximated by circular arcs near enough to give satisfactory results, and I worked out a scheme whereby the machine was modified to form the faces and flanks each by two tangent arcs instead of one. Some improvement in action resulted from this change, but the gearing produced was still noisy, and this improvement was followed by the very radical one of building a cutter forming machine in which the shapes produced were actually rolled. For some time thereafter Wm. Sellers & Company, with whom I was connected, continued to use cylindrical gearing made by cutters of the true shape, but the well known objection to this form of tooth began to be felt, and possibly twenty-five years ago my attention was turned to the advantages of an involute system. The involute systems in use at that time were the one here described as standard, having 14½ deg. obliquity, and another recommended by Willis having an obliquity of 15 deg. Neither of these satisfied the requirements of an interchangeable system, and with some hesitation I recommended a 20 deg. system, which was adopted by Wm. Sellers & Company and has worked to their satisfaction ever since. I did not at that time have quite the courage of my convictions that the obliquity should be 22½ deg. or one-fourth of a right angle. Possibly, however, the obliquity of 20 deg. may still be justified, by reducing the addendum from a value of one to some fraction thereof, but I would not undertake at this time to say which of the two methods I would prefer. In a general way I think Mr. Flanders has raised a very important question, and one which should be taken up by the Society.

3 I brought up the same question nine years ago before the Engineers' Club of Philadelphia, and said at that time that a committee ought to be appointed to investigate and report on an interchangeable system of gearing. We have an interchangeable system of screw threads, of which everybody knows the advantage, and there is no reason why we should not have a standard system of gearing, so that any gear of a given pitch will run with any other gear of the same pitch.

4 I would therefore propose, as the author suggests, that this subject be referred to a Committee of the Society to investigate and to report upon the adoption of a standard system of involute gearing. The paper considers gears from a 12-tooth pinion to a rack only, and that is as far as I would go with such a system. If internal gears are employed, they would necessarily have to be more or less special. I think Mr. Flanders has covered the subject in a very clear and concise

way. He has done a great deal towards the solution of the question by properly stating it, and when anything is properly stated, it is half solved. I therefore make this a motion. (The motion was considered later in the discussion.—EDITOR.)

MR. LUTHER D. BURLINGAME I can easily believe that my friend, the author of this paper, found the solution of the interchangeable gear tooth problem a task far greater than he had anticipated. While many writers appear to reach some rather definite conclusions, I believe that the usual experience of investigators along these lines has been voiced by Mr. Fred J. Miller in the American Machinist, "I think that it is the experience of most men that the more they have studied on the matter of tooth-gearing, the more clearly it has appeared to them that they would never be able to believe anything in regard to it." I take this opportunity to express my appreciation of the able and fair-minded way in which Mr. Flanders has dealt with so difficult a subject, and one that can be viewed from so many points.

2 Mr. Flanders bases his data for the 14½-deg. pressure-angle entirely on a form of tooth which is a true involute for its entire length. As such a tooth is not made or recommended by any manufacturers, as far as I know, and as what is made would give a radically different showing in the comparative tables of the paper, this seems to be setting up a "straw man" to knock over, rather than "tackling" the real thing. It can be said in extenuation that the author used the data at hand; that the data in commercial use were not available is not surprising, as they have been derived by manufacturers through years of experience and at great expense; furthermore, the giving out of such data, even were it good policy, most probably would never result in the production of good gears, much less of interchangeable gears. The experience of the company with which I am connected is that the old saying, "a little knowledge is a dangerous thing," is most applicable to the science and practice of gearing.

3 In illustrating the difference between the theoretical and the commercial tooth of 14½-deg. pressure-angle, reference is made to Par. 7, 10 to 19 inclusive, 33, 37, and 38, and to Fig. 5, 8, 12, 13, and 14, as well as to the table accompanying Par. 45, Comparison of Selected Examples of Involute Gear Tooth Systems. These deal to a greater or lesser extent with the question of length of contact of the engaging teeth and with the question of the number of teeth in

continuous action. To show the radical difference between the results obtained in the paper and based on the use of the uncorrected involute form of tooth with 14½-deg. pressure-angle, and the results actually obtained with cutters made by the Brown and Sharpe Mfg. Co., I would refer to Fig. 1 to 3. As a matter of fact there are two teeth driving for half of the time, or on a basis of Mr. Flanders' table referred to above, 1.5 teeth in continuous action. As his table gives 0.98 of a tooth in continuous action, the real Brown and Sharpe tooth

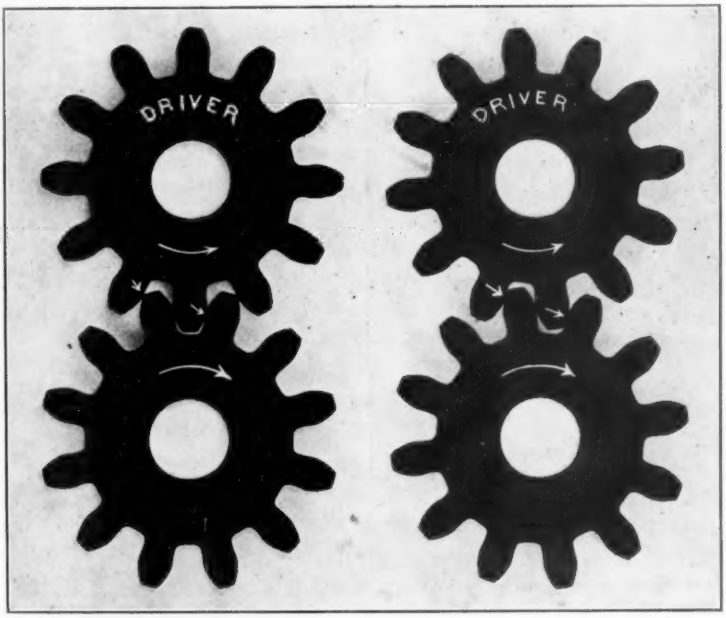

Fig. 1
Twelve-Tooth Pinions B. & S.

Fig. 2
Interchangeable 14½-deg. System

shows a gain of more than 50 per cent above the results tabulated as the Brown and Sharpe standard. If all of the statements in paragraphs and figures above referred to should be modified to this extent, they would give a real comparison instead of a hypothetical one. Thus referring to Fig. 14, Case 1, the diagram would be radically modified for the Brown and Sharpe system where at least two teeth are driving for half of the time, the commercial tooth approaching nearer to what is shown in Case 5 of this same figure.

4 The teeth of the gears cut with Brown and Sharpe cutters are

slightly eased off at their points so as to come gradually into engagement, thus insuring smooth and quiet running. Experience has shown that such a modification is not only important but essential, and in any system, no matter what the pressure angle and height of addendum, I believe the teeth should be so modified. When such a modification is made it becomes a mere academic contention whether the corrected part is modified from a true involute or something else.

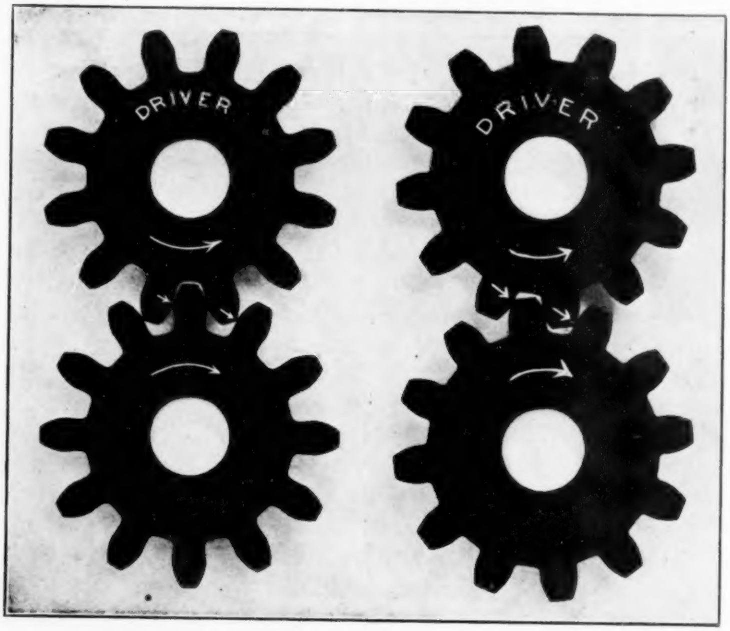

Fig. 3
Brown and Sharpe Interchangeable $14\frac{1}{2}$-deg. System

Fig. 4
Pressure Angle 20 deg.
$$S = \frac{0.7}{P}$$

I understand there are methods covered by patents for accomplishing a rounding and easing off of the points of the teeth when they are formed by the generating process.[1]

5 The impression is given by the author that there are difficulties in the way of using the generating process which limit it and that the devising of a new system of gearing will make its use more satisfactory. This seems like a case where, as the mountain will not come

[1] Bilgram Patents, No. 749 606 and No. 749 683

to Mohammed, Mohammed must go to the mountain. To consider the adoption of one system for gears made by the generating process and another for those made by the use of formed cutters, would be like adopting the metric sytem for calculations and the English system for the use of the workman, considering each as adapted respecttively to these particular uses.

6 In the Brown and Sharpe practice, to meet their own needs and those of their customers, cutters and gears are made with pressure angles varying from 4 to 28 deg., and with a height of tooth ranging in both directions far outside of the limits discussed in Mr. Flanders' paper. It is an everyday matter to produce gears and cutters with such variations, and they can be made with at least equal facility, as compared with standard shapes, by the use of formed cutters. Such cutters, as made by Brown and Sharpe for 20 and 22½-deg. pressure-angle, are for each angle interchangeable.

7 Mr. Flanders has given little consideration to the question of *backlash,* in fact has not mentioned it in Par. 45, where he sums up the objections to using a greater pressure-angle and less addendum. In many classes of work this becomes an important consideration and any system tending to increase backlash is objectionable. It is a common practice in making special gears for printing presses and other places where backlash must be reduced to a minimum, especially where the center distance of the gears must vary appreciably, to make the pressure-angle as low as 4 deg. and to increase the addendum to a greater length than standard. In any case, the greater the pressure-angle the more backlash there will be with a given inaccuracy in center distance, a given variation in setting the cutter or tool when producing the gear, or a given amount of wear on the teeth.

8 There are indeed special cases where a greater pressure angle than 14½ deg. is sufficiently desirable for various reasons to off-set the objections made. The Brown and Sharpe Company use on their machines gears with a greater pressure-angle, when the conditions make it seem desirable, and they make such gears and cutters for customers whenever called for. These, however, are invariably made with a correction for smooth and quiet running, even when the pressure angle and height of tooth are such that theoretically this would not be required.

9 The Reinecker generating machine has provision for easing off the points of the gears to prevent noise.

10 Frank Burgess of the Boston Gear Works says[1] regarding the height of tooth, "A long tooth usually gives a better movement than

[1] American Machinist, June 27, 1907, p. 935.

a short stubby one. With the shorter tooth, the pitch is proportionately greater for its depth and there is a tendency to jump from one to the other, especially for a pinion with less than 20 teeth, and this tendency results in noise. The noise in gearing is undoubtedly the result of shocks, jumps and vibration caused by teeth coming into and going out of action."

10 While the use of shorter teeth or a finer pitch would theoretically make some saving in time of cutting, we do not find such a saving appreciable.

11 I would ask the author why the possibilities of inaccuracy mentioned in Par. 30 as applying to the use of formed cutters are not also present in the generating method, instead of being limited as stated in Par. 31. It would seem to me that most of these possibilities of inaccuracy would be equally present in both systems.

12 With all our experience at the Brown and Sharpe works, our experts feel that the subject of gearing is full of pitfalls, and the more experience we have, the less we feel like dogmatizing or appearing as authorities. While all theories should be examined with an open mind, I believe that a spirit of conservatism should govern their investigation until they are proved by practical experience to be correct.

13 I do not understand that the author suggests an abandonment of the present system, but states rather that the "discussion points clearly to the wisdom of an alternative gear tooth standard of shorter addendum and increased pressure angle." The fact is that modified forms of teeth, of a sort approved by the author and many other forms also, are now used in special cases where they prove to be better. Every manufacturer of gear-cutters knows this to be true, though at the present time, with the large demand for gears in automobile construction, there may be an emphasis upon a tooth of a certain form. What this emphasis will be in future years is uncertain. As likely as not another pressure-angle and another form of tooth may be insisted upon. Is it not better to leave these matters to the manufacturers of gear-cutters and gear-cutting machines who are willing to give the public what they want rather than attempt to formulate a system which the experience of the next few years may possibly render of little value?

Mr. D. F. Nisbet[1] I agree fully with the author's conclusion that an alternative standard of involute teeth for heavy, slow-speed gearing

[1] D. F. Nisbet, Engineer, Pittsburg, Pa.

is desirable; particularly so for such gearing as rolling-mill pinions, where, owing to the limited diameter and tremendous shocks, it is necessary to use few teeth of large pitch, and obtain a comparatively smooth action by making the gears either double helical or of the staggered tooth type.

2 Smoothness of action, as understood and defined by builders of the finer classes of machinery, is not essential for rolling-mill work; perhaps continuity of action would better express the desideratum for this class of work. All that is necessary is a degree of smoothness of action that will not leave traces of "harsh gearing" on the finished product. This applies with particular force to plate mills and sheet mills, and in a lesser degree to all types.

MR. CHARLES WALLACE HUNT It should be borne in mind that this paper treats of a system of interchangeable gears; that is, all of

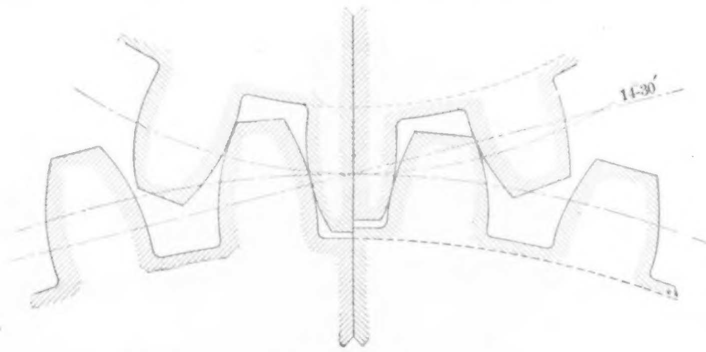

FIG. 1 DIAGRAM SHOWING LONG AND SHORT TEETH WITH THE PITCH LINE IN COMMON

the pinions and gears of each series having the same addendum. This commercial requirement eliminates the consideration of theoretically perfect forms of conjugating teeth for special conditions and confines the discussion to what Mr. Wilfred Lewis designates as the "best compromise between conflicting conditions."

2 Perfectly formed teeth are seldom made and cannot long be maintained owing to wear of the tooth faces. The chief disturbing factors are the shop errors in the pitch diameters of the gears and in the shaft center distances. Such construction errors are unavoidable and so must be tolerated. Some factors partially offset each other, such as the tension caused by the friction of the sliding sur-

faces of the tooth after passing the center, which weakens the tooth. The fillet at the root of the tooth strengthens it. Usually the favorable and the adverse factors partially balance each other, but the working formulae must provide for the unexpected which, as Prof. Sweet reminds us, frequently happens. After considering and valuing the various factors that affect the problem, the designer will decide on proportions that, in his judgment, will give the best general result in practice.

3 In the gear work of the C. W. Hunt Company it is assumed that

The angle of action is 14½ deg.

The whole load is carried on one tooth.

A pinion with less than 19 teeth should not willingly be used.

If the pinion is strong enough its conjugating wheel is also.

A table is quicker and safer to use than a formula.

A table based on fibre stress is preferable to one based on the names of materials.

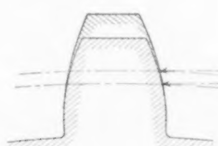

FIG. 2 DIAGRAM SHOWING A SHORT TOOTH SUPERIMPOSED ON A LONG TOOTH WITH THE ROOT-LINES IN COMMON

4 The total length of teeth frequently referred to, expressed as a percentage of the circular pitch, is

Rankine	0.75
Sir Wm. Fairbairn	0.70
Brown and Sharpe	0.687
Molesworth	0.66
Coleman Sellers	0.65
Hunt	0.55

5 After ten years' use in a wide range of machinery, I believe that for commercial machinery the following proportions for cut teeth are especially suitable ones. If they are not ideal proportions the difference is slight. The face of the tooth is involute in form, the angle of action is 14½ deg., and the length of the tooth is 0.55 of the circular pitch.

HUNT TOOTH FORMULA

Addendum	0.25	of the circular pitch.
Dedendum	0.25	" " " "
Clearance	0.05	" " " "
Length of tooth	0.55	" " " "

TABLE 1 WORKING LOADS
SPUR GEARS, ONE INCH FACE

Diametral pitch	1	1¼	1½	2	2½	3	3½	4	5	6	7	8
Circular pitch	3.14	2.51	2.1	1.57	1.25	1.04	.897	.785	.628	.524	.449	.393
Pounds on the pitch line	2250	1800	1500	1100	900	700	600	565	450	375	300	282

6 Table 1 gives the working strength of a spur tooth, having parallel flanks. The tabular load applied on the pitch line produces

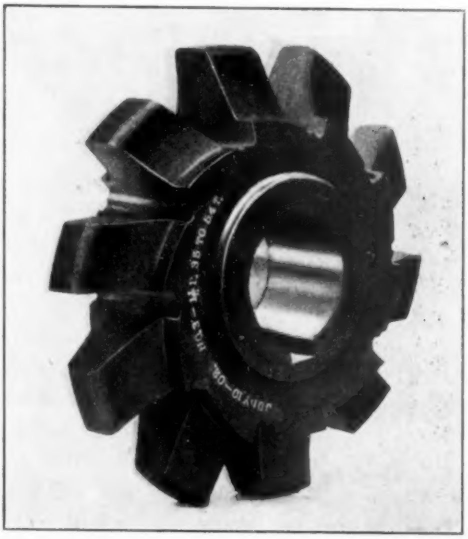

FIG. 3 PERSPECTIVE VIEW OF THE CUTTER

a fibre stress at the root of the tooth of 5000 lb. per square inch. If applied at the pitch line plus one-half of the addendum then the fibre stress will be 7100 lb.; if at the extreme end of the tooth 9200 lb. per square inch.

7 For a safety factor of 3½ when the load is applied at the end of the tooth:

Cast iron (16 000 lb. ultimate strength), use a working load of one-half the tabular number.

Steel (64 000 lb. ultimate strength), use twice the tabular number.

GEAR CUTTERS

8 The initial stresses in the casting frequently cause a distortion when the gear is finished. A spoked gear-wheel blank may be turned perfectly round in a lathe, but after the stocking cutter has been used the points of the teeth are seldom in a true circle. The rim between the spokes either bulges out, or falls in, depending on the initial stress in the metal. This is frequently a material amount and is objectionable as it alters the pitch line and the tooth spacing, and also injuriously affects the clearance.

9 It has been found advantageous to have tooth cutters (Fig. 4) with a wing on each side which will trim the length of the teeth to the standard size. With these cutters a gear made of metal without a hard scale can be cut without sizing the blanks in a lathe.

10 Cutters for teeth proportioned by the above formulae are made for any one desiring them, by Brown and Sharpe, Gould and Eberhardt, or any other gear-cutter maker.

Mr. Oberlin Smith The suggestion has been made that a committee be appointed to standardize gear teeth, with the assumption that there would be one standard. Another has said that it is best to leave it to the makers, as they know the needs of their customers; the result of this would be forty, or fifty or even a hundred standards, as we have now. It would seem wise to standardize the teeth, even if there have to be several kinds. We need to do this almost as much as we needed to standardize the screw thread. It would be useful to have some standard which most people would follow, and undoubtedly in this way a standard practice could be established. The committee would consider the different conditions, as for instance, those met in heavy rolling-mill work, and choose different pressure-angles adapted to each case.

2 If a committee is appointed to consider this matter, I hope they will take out fractions and make the angle of obliquity 15 deg. rather than 14.5 deg. Suppose they found 15 deg. good for a certain class of gears, and 20 deg. good for another class, and 25 deg. good for another class, why could we not have two or three standards, under certain names, so that people could use them to comply with the conditions?

3 If we do have a committee on gearing, their investigations should go further than merely determining shape. It would be well to instruct them to find out all they can about strength of gears as well. The matter of strength is very important, and it is a thing which we do not know very much about. Many of us go by the Lewis formula, by which, it seems to me, the required strength increases too rapidly with increase in speed. But there are conditions of impact and percussion which cause the teeth to vibrate and make them break at high speed. We do not know what these conditions are, and that is what the committee should find out.

4 One thing desirable, in interchangeable gearing, is to be able to use fewer teeth in pinions. Some one said that nine is the lowest number desirable. The Brown & Sharpe Mfg. Co. give twelve as the least practical number to get a theoretically good wheel, and very likely that is so. I have used pinions with four, for some 15 or 20 years, without perceptible wear where the pressures were not great but the speeds were very high, the small pinion being the driver. I have some lathes which have five-tooth pinions doing the driving; they are still in use and are a perfect success. Of course, it is probable that the angular motion is somewhat irregular with such a small pinion.

5 It is often practicable to use a pinion with from five to ten teeth, as a driving but not as a driven gear. It is well to avoid a train of several gears by using only two, having the pinion as small and gear as large as possible. It is often good practice to cut such a pinion in the solid shaft. We are building some heavy presses, of 1000 tons capacity, where the pinions are more than a foot in diameter, forged integral with the shaft, one at each end. A common practice in rolling-mills is to use two helical pinions separately keyed on, with pitch reversed to avoid end-thrust and ends abutting. I recently built a mill for a foreign government, with two 16-tooth pinions forged solid with their shafts, where two projecting collars served as blanks for the pinions, and by the simple expedient of separating them, left room for a Brown & Sharpe cutter to run through, into the space between.

Prof. W. Rautenstrauch It seems as if there were sufficient interest in the principles of machine manufacture to justify the formation of a section to consider papers of this kind and draw fire along this line. I have had the matter in mind for some time, and have talked with other members, and the idea seems to meet with approval. I therefore

propose that we meet this afternoon to consider the formation of such a section, and draw up a letter petitioning the council to form such a section immediately. (The meeting was held and a committee appointed to formulate plans for a machine shop section to be submitted to the council and reported at an adjourned meeting at the time of the spring meeting of the Society.—Editor.)

Mr. Charles H. Logue[1] After reading Mr. Flanders' paper, it would be useless for me to enter a discussion as to the need of a universally accepted standard for gear teeth, especially for large gears. It is now common practice for all users and makers of large gears, to correct the teeth for interference, either by shortening the addendum or increasing the angle of obliquity as compared with the 14½ deg. system, or both. This procedure, of course, entails additional expense in manufacture, as special cutters and formers must be made for each gear. Again, each manufacturer, and in many cases the individual user, has his own ideas as to how this correction for interference should be made, and also as to the amount.

2 I wish to express my approval of the purpose of Mr. Flanders' paper, namely: the appointment of a special committee of the Society to recommend an interchangeable gear-tooth standard, and also of the action at the last meeting of the Society whereby a recommendation was made to the Council that such a committee be appointed. The present situation is little short of chaotic, and the present "so-called" standard was most aptly characterized by Mr. Wilfred Lewis as "no standard at all." The work before this committee is mountainous, for this problem is probably more complex than any other individual problem in machine design and construction that could be presented. The difficulty of the problem is also a measure of the benefit that will come to the machine-building industry if a standard is recommended and brought into general use.

3 It is apparent that there will be a great saving in the adoption of a rational, universal system, as well as an improvement in tooth-action, the extent of which is entirely unappreciated by gear users. Only a comparatively small part of the gear-tooth surface is in actual use, and the rest of this surface is a detriment rather than a help to the tooth-action. In addition to this detriment, the correction as sometimes made produces surfaces that give an irregular impulse, or series of impulses, to the driven gear by the driver. Thus the teeth of the driven gear have instantaneous accelerations and retarda-

[1] Charles H. Logue, with R. D. Nuttall Co., Pittsburg, Pa.

tions and lack a regular motion. This, of course, tends to destroy the gears rapidly. I believe that this condition prevails to a greater or less extent, in all involute gears as ordinarily made.

PROPOSED SHORT TOOTH STANDARD

4 In order to set before the Society information as to existing gear-tooth systems, I propose to describe a short-toothed, 20-deg. standard that I began to use some three years ago. Since that time I have advocated this system for all gears heavier than one-diametral pitch, and in many cases for gears of finer pitches. I now see no reason why this system is not universally adaptable. It follows closely the Fellows system as worked out for fine pitches, except that the addendum bears a definite relation to the circular pitch. This relation is expressed by the equation,

$$Addendum = 0.25 \ circular \ pitch$$

5 This quantity was derived from experience and from a study of the tooth parts of the Fellows system. In the Fellows system the addendum varies from 0.264 of the circular pitch to 0.226 of the circular pitch. These factors for the common combination pitches are shown in the following tabulation:

For	2/2½	pitch,	addendum =	0.255	of circular	pitch.
"	2½/3	"	"	0.264	"	"
"	3/4	"	"	0.240	"	"
"	4/5	"	"	0.255	"	"
"	5/7	"	"	0.228	"	"
"	7/9	"	"	0.247	"	"
"	9/10	"	"	0.254	"	"
"	10/12	"	"	0.264	"	"
"	12/14	"	"	0.226	"	"
"	14/18	"	"	0.250	"	"

6 The factor 0.25 is, therefore, a rough mean of the factors shown in the table, and is also a convenient quantity to use in computing gear-tooth parts. This latter advantage is especially true as the thickness of the teeth at the pitch-line, added to the pitch diameter, gives the outside diameter, and this relationship is of convenience when measuring gears in which the pitch-diameter and pitch are unknown. In many cases the only dimensions received by the gear manufacturer are the outside diameter and number of teeth. While this feature should not be an influence in determining the length of

addendum to be used, it is a convenient point to keep in mind. For diametral-pitch the addendum is found by dividing 0.7854 by the pitch. This is a well-known factor; is equal to $\frac{\pi}{4}$; is in general use in many engineering formules, and is easily memorized.

7 The tooth parts for circular pitch are:

Addendum	0.25 × cir. pitch,	instead of 0.3183
Dedundum	0.30 " " "	" 0.3683
Working depth	0.50 " " "	" 0.6366
Whole depth	0.55 " " "	" 0.6866
Clearance	0.05 " " "	same as now used

8 The tooth parts of diametral pitch are:

Addendum	$\frac{0.7854}{P}$	instead of	$\frac{1}{P}$
Dedundum	$\frac{0.9424}{P}$	"	$\frac{1.57}{P}$
Working depth	$\frac{1.5708}{P}$	"	$\frac{2}{P}$
Whole depth	$\frac{1.7278}{P}$	"	$\frac{2.157}{P}$
Clearance	$\frac{0.157}{P}$	same as now used.	

9 In addition to the length of the addendum the other important element of the gear-tooth, to be determined, is the obliquity. While the increase in journal-friction with the pressure-angle is not as great as is generally supposed, yet this angle should be kept as small as is consistent for the purpose intended; that is, to obtain an interchangeable involute system without correcting the tooth-outlines. In the system which I am describing 20 deg. is the angle of obliquity. Thus the two essential elements of the system are, the addendum equal to 0.25 of the circular pitch and the pressure-angle of 20 deg. These correspond very closely with Case 4, as set forth in Par. 44 and 45 of Mr. Flanders' paper. He characterizes Case 4 as a system having an angle of obliquity of 20 deg. and an addendum height equal to $\frac{0.8}{P}$. For circular pitch the addendum would equal 0.8 or 0.2513 of the circular pitch. That is, Case 4 is essentially the system which I am describing.

10 Referring to Par. 45 of Mr. Flanders' paper, from his tabulation captioned, Comparison of Selected Examples of Involute Gear Tooth Systems, I have transcribed the factors:

Smallest pinion in series, to avoid interference	14
Maximum number of teeth without correction for interference with 12-tooth gear	36
Maximum and minimum number of teeth in continuous contact	1.38 1.18
Proportion of side pressure on bearing to tangential pressure	1.064
Strength-factor of rack	0.543
Strength-factor of 12-tooth gear	0.354
Comparative loss of work from friction	552
Comparative durability	1.44

11 He states that the smallest pinion in the series that can be used and avoid interference is one with 14 teeth. The lowest theoretical number of teeth for such a pinion engaging a rack is 13.4. I believe, however, that 13 teeth can be used without correction as the error is less than ½ a tooth, therefore in order to use a 12-toothed pinion, which is today considered a *desideratum*, the only correction necessary is for this pinion alone. In some practice, street railway use in particular, 14 is the smallest number of teeth which is ever used; 12 and 13 are always avoided.

12 In order to use an uncorrected 12-tooth pinion in this instance, without changing the addendum, it would be necessary to increase the angle according to Mr. Flanders' formula, to 21 deg. 13 min. If it is necessary to include the 12-tooth pinion in the interchangeable system, I would prefer to see the angle increased for the reason which I have indicated above. However, I would strongly recommend for consideration an angle of 20 deg. and the addendum as given, and thus consider a 12 and perhaps a 13-toothed pinion as special. These would then be in the same situation as are 10 and 11-toothed pinions today.

13 It will be argued that the short-tooth standard permits of a smaller number of teeth in contact. While this may be true, in some cases, the superiority of this contact has been demonstrated by Mr. Flanders. However, it should be noted that, owing to the increased strength of the tooth that has been shortened, and has a wider angle of obliquity, the pitch may be reduced accordingly to give the same number of teeth, or more, in contact, as are in contact in similar gears having the usual length of addendum and a pressure angle of 14½ deg.

14 As a matter of interest it must be pointed out that this system is now in use for the gears of the subway trains in New York, and to the best of my knowledge it is superior in service to the standard-toothed gears that have been discarded. The special 20-deg. involute stub-tooth system described by Mr. Litchfield in his paper on Spur-Gearing on Heavy Railway Motor Equipment, is the same as this short-tooth 20-deg. standard that I have proposed for consideration.

Mr. E. R. Fellows The conclusions arrived at by the author, as given in table accompanying Par. 45, do not entirely coincide with the writer's experience. The reason is that the form of 14½-deg. tooth considered is virtually a short tooth of this angle, and has theoretically, in these deductions, some of the advantages of the stub-tooth. The forms selected by the author for the standard or 14½-deg. tooth are those which he infers are produced by a set of formed milling-cutters designed to cut an interchangeable set of gears. As, however, the exact form of these cutters is more or less of a trade secret, he has evidently been obliged to make his own deductions upon some points, such as the exact modification for interference and the angle of the flank below the base line. A little variation in these points makes a considerable difference in theoretical efficiency.

2 The stub-tooth so-called, which is a short tooth of 20-deg., being very little modified for interference, conforms very closely to the theoretical. The standard forms selected give n, the number of teeth in contact, as 0.98. It has been the general experience that gears having a line of action as short as this do not run satisfactorily; they are noisy. And this is the test by which gearing is approved or condemned; a difference of one or two per cent in efficiency being of no importance whatever. An examination of any satisfactory set of gearing of standard form will show that most of the gears bear nearly if not quite to the point. This means that the value of n is ordinarily 1.5 to 1.75, and while the running qualities are satisfactory the efficiency is lower than if the value of n were 0.98.

3 This being the case, it is safe to assume either that the form of standard cutters ordinarily used is not the one discussed by the author, or that gearing, after running a short time, changes from wear sufficiently to give a longer line of action. The author evidently had this condition in mind, as his formula for strength assumes that the stress is applied at the extreme point of the teeth, notwithstanding the fact that according to his first deductions, this would be impossible. We will consider the changes necessary to give to n the value of general practice.

4 In the case of two 12-tooth pinions shown in Fig. 2 and 3, the author assumes that interference begins as soon as the theoretical action ends. This is not the case. The path traced by the point D, Fig. 3, beyond the so-called interference point, is an epitrochoid, which so nearly conforms to the theoretical involute, that if the flank of the tooth below the base line be undercut $2\frac{1}{2}$ deg., D will rub the entire distance *ab* of the point of the meeting tooth. The radial flank causes what little interference there may be.

5 The design of cutters for an interchangeable set is a matter of compromise. Strength demands as broad a flank as is possible for the pinion; efficiency demands a short line of action; quiet running requires a longer one. The latter being the point by which the user tests his gears, it is safe to say that it receives the most consideration. A little compromise would obviate even the undercutting of the flank.

6 Theoretically, in the interchangeable set two pinions of 12 teeth should mesh together. Practice does not demand this, such a case being extremely rare. This combination would give a very low efficiency. If this is imperative, as in the case of spur gear differential of the automobile, gears of greater angle are almost invariably used. It is safe to say that the meshing of two 15 or 16-tooth pinions would fulfill all practical requirements, and anyone who has rolled two standard pinions of 12 teeth together, in the condition in which they leave the cutter, will decide that in practice all gears are not absolutely interchangeable.

7 The writer's experience has been largely with generated gears, but from this experience, he would say, that the best practice in gearing of the standard form demands a length of action giving about $n = 1.5$, and that this requirement is met. This value of n would materially change most of the diagrams given, particularly Fig. 12, comparing the lost work, and Fig. 13, comparing durability. If the modification for interference of the standard 12-tooth pinion, instead of being 0.4 of the addendum, be 0.0, and if this modification up to about 30 teeth be a very immaterial amount, the friction or lost work of the average train of gears will be considerably more than that given by the author.

8 In this connection the term "corrected for interference," used by the author and by other authorities on gearing, is open to criticism. As the involute curve is theoretically correct, and after "correction" is only *correct* in the sense that it will mesh with another form which is not in itself theoretically correct, the term "modified" more nearly describes the case.

9 The writer would emphasize the statement in Par. 37 regarding durability. The fact that wear, in the case of the stub-tooth, is distributed over a much greater surface, is a strong point in its favor. This is more marked in the case of the pinion, where most of the wear usually takes place. The result is, that the form of the worn-out stub-tooth is practically that of an involute curve.

10 The "handicap" of "one form of generating process" is not as serious as might be inferred from Par. 39. As the modification for interference is very simply taken care of by the design of the cutter, it is possible to vary this modification automatically, and to give the teeth of each diameter of gear just the required amount and no more.

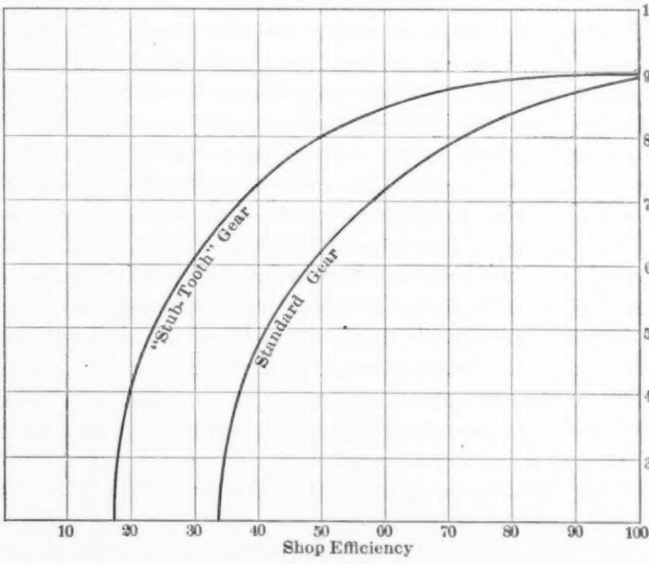

Fig. 1 Comparison of Running Conditions of Standard and Stub-Tooth Gears

11 One important point not touched by the author, is the effect of what may be termed "shop efficiency" upon the comparative running qualities of the standard and the stub-tooth systems. It is unnecessary to argue that under certain conditions the running qualities of the stub-tooth are much superior to those of the standard tooth: these conditions are rare. Fig. 1 is based upon experience, as it is impossible to reduce most of the factors to figures. The vertical lines represent different degrees of shop efficiency, 100 per cent being possible only under laboratory con-

ditions. 90 per cent represents what are probably first-class commercial conditions, and 70 per cent is more common. The horizontal lines represent running quality. 100 per cent would represent the noiseless gear, but 90 per cent is about the highest point attainable, the difference of 10 per cent representing errors impossible to eliminate, such as change of metal after cutting, etc. The two curves would probably very nearly meet under laboratory conditions, but under practical ones the balance is in favor of the stub-tooth.

Mr. Thos. Fawcus[1] We find the Brown and Sharpe system sufficient in every respect up to 1 P. For large gears we generally use the 20-deg. involute, with a depth of 0.55 of the circular pitch, or about 0.8 S. A layout for some 5½ in. pitch gears recently made is shown in Fig. 1.

2 The gear manufacturer is usually in the unfortunate position of having to assume responsibility for gears designed by others for work of which he is not informed. If the gears fail, blame falls on the only detail left to his discretion; namely, the kind of iron or steel or other material used. A recent case where material was blamed for being "too soft," upon investigation showed gears put to a service calling for nearly four times the strength by Lewis' formula. For these reasons I think the question of durability has not received sufficient attention. Tooth forms do not trouble gear manufacturers much; it is easy to make conjugate involute teeth, and, for heavy gears at least, the short tooth is preferable. But we need to know more about the durability of certain combinations of metals at high and low speeds. This is most important in gears under heavy load and at low speed where the material must be such as will not crush on the line of contact; also in worm gearing, which is in a class by itself with regard to the materials of which it should be made. The usual practice of making the worms of steel and the wheels of cast-iron is very bad. But if the gear-maker offers any objection, he is confronted with the fact that the combination has proved satisfactory in some particular case. That results are most varied and even contradictory is true; one pair of worm gears succeeding where a perfectly similar pair will fail. This is sometimes partly explained by excessive friction caused by tight bearings in new machines, or shaft centers too close together, or insufficient lubrication. The cutting action once started is difficult to stop and usually results in the destruction of the wheel.

[1] Mr. Thos. Fawcus, Fawcus Machine Co., Pittsburg, Pa.

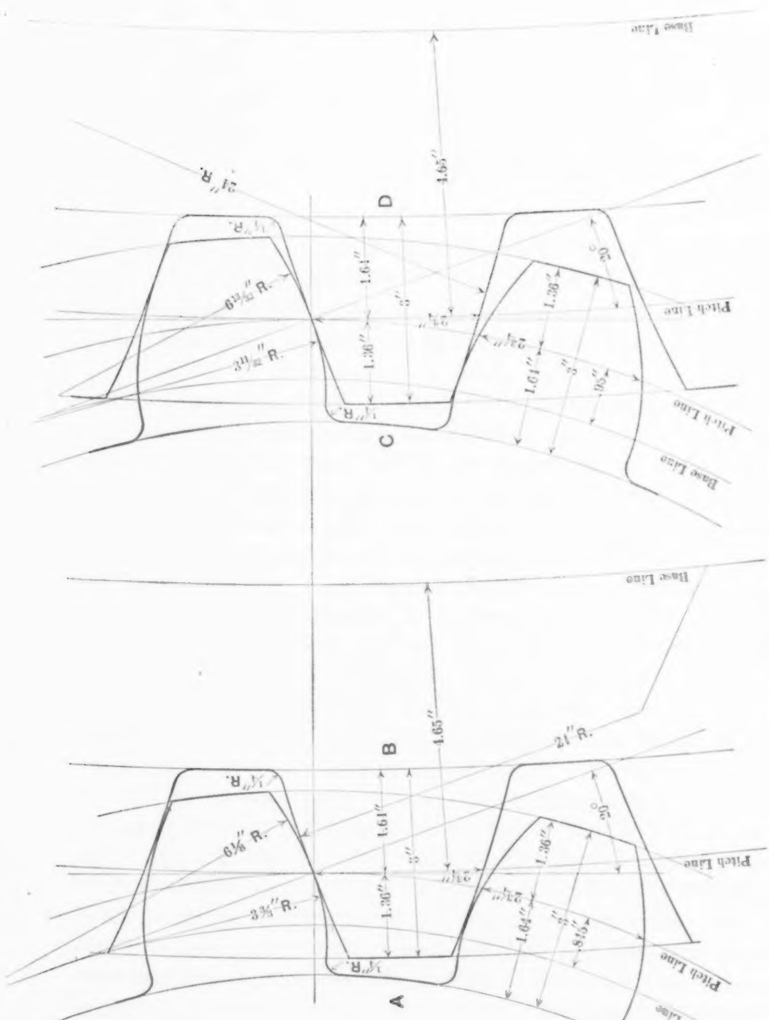

FIG. 1 SPECIAL 20-DEG. TEETH

Pinion A, 16 Teeth; Pitch, 5½-in.; Face, 18-in.; Pitch-Diameter, 28-in.; Radius Base Circle, 13.155. in.
Gear B, 88 Teeth; Pitch, 5½-in.; Face, 18-in.; Pitch-Diameter, 154-in.; Radius Base Circle, 72.35 in.
Pinion C, 18 Teeth; Pitch, 5½-in.; Face, 18-in.; Pitch-Diameter, 31½-in.; Radius Base Circle, 14.8 in.
Gear D, 88 Teeth; Pitch, 5½-in.; Face, 18-in.; Pitch-Diameter, 154-in.; Radius Base Circle, 72.35 in.

3 This is, I know, departing somewhat from the subject under discussion, but I mention it because it is a very real difficulty with which engineers and manufacturers have to contend.

PROF. F. DER. FURMAN At the end of Par. 12 the author states: "In Fig. 7, it will be seen, a similar increase of addendum makes no change in the number of teeth in contact, as the action is still limited by points C and D." In Par. 15 it is stated: "When $\alpha = 20$ deg. and $S = 1.0$, the amount of contact between any two gears of the same series from 12 teeth to a rack is constant at about 1.4" and also "that if in any series there is interference in the case of two pinions having the smallest number of teeth allowed by the series, the amount of action obtained in that case is constant for any other case throughout the whole series, up to that of two racks meshing with each other. As far as the author knows, this condition has never before been noticed." Also in Par. 15 we find: "The 14½-deg. standard series of whatever height of addendum gives less than continuous action, being about 0.987."

2 All the above statements must rest on the assumption that there is no contact between the tooth surfaces beyond the interference line, in which case there is a fair portion at the end of the tooth that could be cut off without interfering in any way with the running. If a set of interchangeable wheels were made with conjugate extensions beyond the interference line none of the statements quoted would hold. Since the extension cannot be involute, and must be of some other form, that form might as well be the simplest one, which is the epicycloidal. And further if we give the epicycloidal form to a fair portion of the face, and flank, and leave only a portion of the tooth near the pitch circle, of involute form, we may as well abandon the involute system entirely for interchangeable gearing, and adopt the epicycloidal system instead for interchangeable work.[1] Then in the endeavor to perfect gearing we could at least work towards theoretical conditions, instead of using the approximate methods referred to in Par. 32 for forming the extensions to the involute beyond the interference line. These extensions I understand are not in practice true conjugate curves and therefore must allow, theoretically at least, the follower to slow up while the driver

[1] To meet the difficulty involved in cutting epicycloidal teeth at the pitch circle, we could adopt a combination outline for interchangeable teeth in which the tooth form is involute for a short distance on either side of the pitch line and epicycloidal for the remainder of the face and the flank.

continues at its uniform velocity, the result being a blow as the teeth come into contact.

3 The author shows that in a system having an angle of $22\frac{1}{2}$ deg. or more, and where $S = 0.8$, there is no interference, and it follows that the teeth may be correctly made of the involute form all the way to their tips. But even here smoothness of action will not be obtained so long as the teeth are cut with standard rotary cutters in which each one must cut a certain range of wheels. Take for example the No. 3 cutter cutting 35 to 54 teeth, it would follow that the cutter must be formed to suit the involute of the smallest wheel and therefore would cut away too much of the face of all wheels having 36 to 54 teeth. This means that there will be correct driving contact only at or near the line of centers, and as the driving pair pass through receding action the follower slows up, thus allowing the next tooth of the uniformly moving driver to come into violent action with impact and noise, instead of easy contact with the follower tooth.

4 In Par. 24, the author states that "In order to get the proper form of fillet on the 12-tooth pinion, the rack tooth was lengthened by an amount equal to the clearance, and the corner of the extended tooth was rounded with a radius equal to $\frac{2}{3}$ of the clearance." I would like to ask why the author goes to the trouble of lengthening the tooth and then rounding it off instead of simply finding the curve traced by the corner of the rack tooth on the revolving plane of the pinion and then placing the fillet curve within the one thus found?

5 From the present paper it would seem that there would always be difficulty, if not impossibility, in producing a series of smooth-running interchangeable gear wheels, all the way from a 12-tooth pinion to a rack, on the involute system. If a series of intermediate sized interchangeable involute wheels were desired the problem would appear to be an easier one.

6 The most difficult part of the problem for interchangeable involute wheels lies in the fact that the face or addendum must be the same throughout the series. It is quite different, however, for an independent pair of wheels which are always to run together. Here the addenda may be made unequal to great advantage, thereby avoiding in most cases the trouble due to interference, and giving a receding action greater than the approaching by any desired amount. This involves the making of a special cutter for each wheel, but where a large number of wheels are to be duplicated the extra expense for cutters would be more than offset by the smoothness due to correct theoretical action.

7 An example of the use of teeth having unequal addenda will be given in a discussion of Mr. Litchfield's paper to be presented this morning. In that illustration it will be shown that a very strong tooth may be obtained with a $14\frac{1}{2}$-deg. pressure angle because: (*a*) the necessary amount of action may be obtained with this angle by a relatively short tooth; (*b*) a specially-formed large fillet may be placed at the root.

8 From the above, and from deductions from Mr. Flanders' paper, it would appear that if we give to high-class gearing the special consideration which each case deserves, theory is pointing to the use of a small pressure-angle for two wheels that are to run always together, and to a large angle for a series of interchangeable wheels, if the involute system is used.

Mr. A. L. DeLeeuw The main point which has been made perfectly clear to me is that there is some difference of opinion on this subject, which is of some significance as showing that many of us have reached a point where we are striving for something different, supposing it to be better.

2 The motion by Mr. Lewis is to the effect that the Council be requested to appoint a committee to consider a standard system of gearing. I am inclined to think this motion should be broadened by making it read "standard systems" instead of "a standard system." Not necessarily that we wish more than one system, but that it would leave the committee which investigates the matter free to consider it from all possible angles. I will therefore second the motion of Mr. Lewis and ask him kindly to assent to my suggestion to broaden the terminology slightly so as to include any kind of system or systems which may be proposed.

Mr. Lewis accepted this suggestion, with the proviso that the resolution should be confined to involute interchangeable gearing; and also a suggestion by Mr. E. H. Neff that it be put in the form of a recommendation to the Council that they take action upon its provisions. The motion, which is expressed by the following, was then put and unanimously carried.

2 Resolved: That the Council be asked to appoint a committee to investigate the subject of interchangeable involute gearing and recommend a standard, or standards, if found desirable.

The Author Referring to Par. 2 of Mr. Burlingame's discussion, it should be noted that I did *not* base my data on a form of tooth

which is a true involute for its whole length. In fact, a main point of the investigation was to show how little of the outline (about one-third) could, in the standard form, be involute. This is one of the grounds for criticism of the present form of tooth, which thus relinquishes, to a great extent, the advantage which the involute curve gives, of perfect action at varying center-distances. Besides, the small length of theoretical curve possible makes the form of tooth indeterminate, except by empirical methods. I followed an old suggestion in making the indeterminate portions of cycloidal form; this was done only in order to obtain a reasonable working-basis from which to start the investigation, and was made necessary by the lack of definite information from the proprietors of the present system.

2 In Par. 3, Mr. Burlingame objects to basing the calculations for the number of teeth in contact, etc., on the small amount of theoretical action possible with this hypothetical tooth form. This procedure is justifiable, since, in a partially involute form of tooth, only the involute portion remains in action if the center-distances (as must be expected to happen in practice) are slightly greater than are called for theoretically.

3 It is true, as intimated by Mr. Burlingame in Par. 7, that the smaller the pressure angle, the smaller the backlash when the gears are slightly separated. This should have been included as one of the advantages of the present system for such uses as change-gears, printing-press work, etc.

4 I do not wish to be put in the position of discounting the complexity of the problem of interchangeable gearing. This question is indeed one of some difficulty. I do feel, however, that a new solution may be safely sought, through the ability and experience at the command of the Society. The fact that the Brown & Sharpe standard gives excellent results for a wide field of work, even when the whole range from 12 teeth to the rack is covered by a set of 8, or at the most, 15 cutters, would indicate that the refinements hinted at are more imaginary than real; and the fact that two very different systems of interchangeable gearing (those mentioned by Messrs. Lewis and Hunt) have been tried out by exacting and competent engineers over a long period of years and over a wide range of application, to the entire satisfaction of the users, would indicate that departures from the old form can be made without fear of disastrous results. Furthermore, there is no practical engineering problem which cannot be solved by the combined application of technical training, perseverance, and common sense.

5 Mr. Fellows (Par. 1) makes the criticism that a longer theoretical action would have been obtained if the 12-tooth pinion had been considered as having the involute extend out to the points of the teeth, and if the flanks of the mating teeth had been undercut, when necessary, to allow this. It is true that more action might be obtained by a system so designed; this action would be truly involute in large gears, but in the case of small pinions it would amount, practically, to a rocking of the face of the tooth about the base of the involute on the mating tooth. This action is purely fortuitous, and is not susceptible of analysis. Besides, it undercuts the flanks of small pinions, leaving a distinct shoulder at the base line. This shoulder does not appear in teeth shaped by standard-formed cutters, where the involute merges smoothly into what appears (on the 12-tooth pinion) to be practically a radial flank. Thus no escape is left from the conclusion that teeth shaped by standard-formed cutters depart at their points very materially from the true involute form. On the other hand, if my memory serves me, a 12-tooth pinion generated by Mr. Fellows' process shows a distinct shoulder at the base circle, so it is very likely that the involute in that case extends nearly or quite to the points of the teeth.

6 Professor Furman calls attention to the fact that the cycloidal system avoids all the corrections and modifications necessary with the involute system; and he suggests that its claims be considered in place of the involute for use in a standard system. It is true that the cycloidal form shows a marked advantage from the standpoint of pure kinematics; but, as Mr. F. J. Miller has pointed out, natural evolution, in free competition with the involute form, has resulted in the practical elimination of the cycloidal system. One of the disadvantages of the latter is the difficulty of obtaining sufficient side clearance for formed cutters; another is the difficulty of generating the curves as compared with the involute. For a general-purpose system of interchangeable gears, the cycloidal form is "out of the running."

7 Professor Furman, in Par. 4, asks why I used the method described for generating the flanks and fillets of the teeth used in finding the strength-factors. The most convenient way to study a standard system theoretically is to consider it as generated from a standard rack. This has its counterpart in actual practice in the action of the gear-hobbing machine. To obtain, then, a rationally practical form of tooth, it was considered to be generated from a hob. In making a hob, I would round the corners as described, to obviate

a rough or stepped generation of the fillet; this procedure also makes the fillet as large as it can be, and still be safe from interference with a sharp-cornered rack tooth.

8 His suggestion in Par., 6 that much better results can be obtained by departing from the regulation proportions, is true. I have given much study to this matter. Many cases are found where the added expense of special cutters is warranted. This use of special gearing is becoming more common than many designers realize, especially in such work as printing-presses, metal-planers, etc., where the best possible results are required. I wish to make the suggestion that all such special gearing be stamped with some recognized symbol to show that it is special. This would avoid costly errors later, when the inevitable repairs have to be made, since it is often impossible to tell by inspection, or by any ordinary means of measurement, whether or not a gear is of standard form. All this is aside from the question at issue, however.

9 Mr. Logue's discussion expresses the viewpoint of the engineer who has specialized in the design and manufacture of gearing, and very logically and forcibly describes present conditions in the field of heavy work. These conditions now appear to be well on the road to remedy, thanks to the action of the members and the Council of the Society.

No. 1219

SPUR GEARING ON HEAVY RAILWAY MOTOR EQUIPMENTS

By Norman Litchfield,[1] New York
Non-Member

In the operation of ordinary street railway motors the gearing is not a serious factor on account of the low horse power required and it is not until the equipment becomes similar in proportions to that of a steam railroad with congested traffic that its importance begins to be felt. With the installation of electric train service on the Manhattan Elevated Railway in 1901, however, the gearing question forced itself on the attention of the engineers through the large number of failures immediately occurring. On the New York Subway equipments the breakages were still more numerous, for these trains are more powerful than any heretofore used, an eight-car subway express train having motors aggregating 2000 h.p., equivalent to a locomotive of about the same power as the new electric locomotives of the New York Central Lines, but differing from the latter in that all the power is transmitted by gears, while the Central's locomotives are gearless.

2 It is evident that the greatest work done by the gearing on any locomotive is during the period of acceleration, and that for a given mileage the total amount of work done will depend upon the rate of acceleration, weight accelerated per gear, and total number of such accelerations or starts. During the evening rush hours in the New York Subway, the total load per gear is about 35 tons, and this weight has to be accelerated at the rate of 1.25 miles per hour per second every third of a mile, that being about the average distance between stations.

3 How serious a matter the breakage of one gear in the rush hour becomes will at once be realized when we state that at one of the many stub end terminals of the system 37 trains in and out per hour are handled during the period of maximum traffic. For the purpose of comparison the train movement at the Grand Central Station of

[1] Engineer Car Equipment, Interborough Rapid Transit Co.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

the New York Central Railroad may be cited, the maximum number of trains there being 35 per hour, and this is, we believe, considered rather extraordinary congestion. One such breakage in the evening rush hour therefore means an inevitable delay to service of half an hour or more and the disruption of the train schedule for the rest of the day, thus entailing great discomfort to the traveling public.

4 Both the elevated and subway trains of New York are operated by the Interborough Rapid Transit Company and there is no doubt that the conditions under which their trains operate are unique, further evidence being furnished by the failure of various apparatus other than gears which had proved entirely successful on other roads. We believe therefore that the data obtained by this company may be taken as measures of the limitation of power transmission by gearing in railway service.

5 Extended studies which we have recently made tend to show that the question of strength has not heretofore received the attention it merited, but that the line of progress followed most sedulously until very recently has been that of increasing the resistance to wear. That this is a highly desirable object cannot be denied, but our experience with the Interborough equipments has shown the most pressing need to be to overcome absolute failure.

6 It is this phase of the question (absolute failure) which we have felt would be of interest to the Society, and so present in this paper the data which we have brought together on the subject.

7 As is probably familiar to all, the electrical operation of the Manhattan elevated was begun some three years previous to the opening of the New York subway, and our earlier experience was therefore with the equipment for the former. The initial gearing installation on the 125 h.p. motors consisted of wrought steel pinions and solid cast steel gears of 3-diametral pitch, this pitch being adopted on account of the economical current consumption thereby obtained. The pinions at once began to fail at the rate of about 15 per month. The failures continuing, it was decided to withdraw all of the gearing then in service and replace it with 2½ pitch on account of the greater tooth section thereby obtained, although, as before stated, this meant some loss in economy of current consumption. This change practically ended the failure of the pinions, but not entirely of the cast gears, and it was decided that greater reliability could be obtained by adopting a composite type of gear consisting of a cast steel center on which a wrought steel rim was shrunk. This combination of a wrought steel pinion and wrought rim gear of 2½ pitch has proved generally

satisfactory for the elevated service, and the improvement to be looked for is therefore in the line of greater wearing life.

8 On the subway division the motors are of 200 h.p. each and the original gearing equipment consisted of solid cast steel gears with wrought steel pinions, diametral pitch 2½, and teeth of the Brown and Sharpe standard 14½ deg. involute. As on the Manhattan division, so on the subway, it soon became evident that the design was not proper, but in contrast to the elevated it was the gears which first caused the trouble. The cast steel gears, therefore, were all scrapped and replaced by the wrought rim type. (It should be stated that the idea of these composite gears was induced by the experience of the Interborough Company, although they were first brought out in connection with the proposed electric motor car equipments for the New York Central lines.)

9 This improvement practically ended the gear breakage, but unfortunately the pinions began to go, the breakages averaging over one a day, and furthermore we find it unsafe to run a pinion the teeth of which measure less than $\frac{3}{16}$ in. at the top. We are therefore compelled to scrap material which should be available for wear.

10 To what then, shall we look as a remedy? Three suggestions have been advanced by the gear manufacturers:

a Diametral pitch less than 2½.

b Steel with elastic limit of 90 000 lb. per sq. in. and over, as compared with our present 45 000.

c 20 deg. stub teeth.

11 The first suggestion we are unable to accept on account of some local conditions, but the combination of the other two seems to have possibilities of success, and we are now replacing all our gearing with specially treated carbon steel with stub teeth.

12 Let us now consider the loads the teeth have to carry and determine why it is that we have to resort to the use of special designs and material in order to make our gearing stand up.

13 Fig. 1 illustrates an express run in the subway, giving the power consumption per train of eight cars, five of which are motor cars, each carrying two motors, thus making ten gears per train. On the same diagram curves are plotted showing the fibre stress in a standard 14½ deg. involute and a special 20 deg. involute tooth both worn to our present limit of $\frac{3}{16}$ in. at top of tooth, the tooth outlines being shown in Fig. 2. The force acting at the pitch line of the tooth has been figured from the power consumption curve and the motor

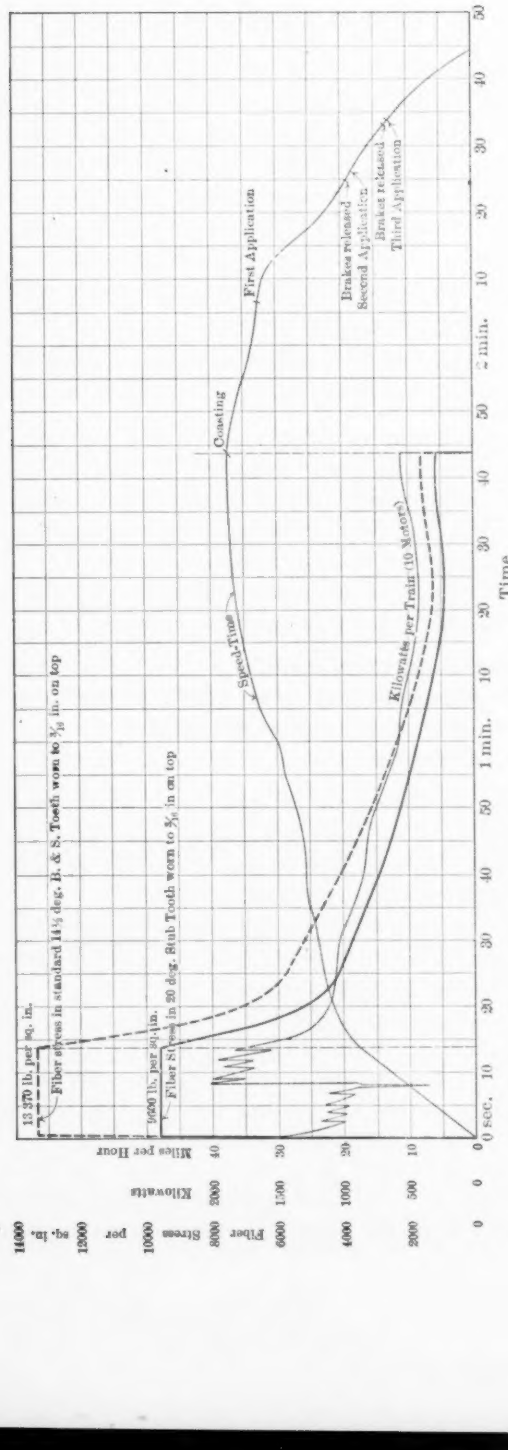

FIG. 1 STRAINS IN TEETH DURING EXPRESS RUN—SUBWAY DIVISION (CALCULATED BY LEWIS METHOD)

characteristics, and the fibre stress in the tooth has then been calculated by the method outlined by Mr. Wilfred Lewis in his paper before the Philadelphia Engineers' Club in 1893, which consists in assuming that all the load is carried by one tooth over its entire face, but at the extreme top, so that the tooth is considered as a beam loaded at one end (the crest) and supported at the other (the base). He states that the force which can be safely transmitted lessens as

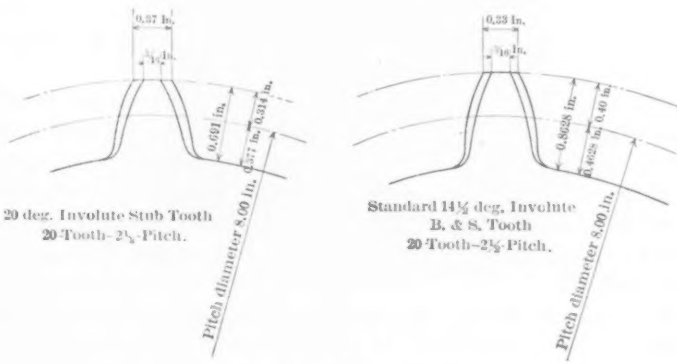

FIG. 2 OUTLINES OF NEW AND WORN TEETH

the speed is increased and gives figures which may be used as safe working stresses at various speeds. These values when plotted give a curve which may be closely represented by the equation

$$S' = S \frac{600}{600 + V}$$

in which

S = safe working stress in pounds per square inch at no velocity.

S' = Safe working stress in pounds per square inch at velocity, feet per minute.

V = velocity of pitch circle in feet per minute.

14 Referring to the diagram it will be noted that the worst condition occurs just at the commencement of the motor curve, this being the point at which the gearing is transmitting the maximum torque at the greatest speed obtained during the period of acceleration, the maximum torque occurring of course, only during this period. Under the conditions shown the fibre stress at this point is 13 370

lb. per sq. in. in the standard $14\frac{1}{2}$ deg. involute tooth, and at this instant the train is running at 17.3 miles per hour, with a corresponding gear speed of 1168 ft. per minute. On account of this speed, therefore, we require a material which will permit the use of a safe working stress at no speed,

$$S = 13\ 370 \times \frac{600 + 1168}{600} = 39\ 400 \text{ lb. per sq. in.}$$

The elastic limit of the material we have had heretofore runs about 45 000 lb. per sq. in. and we therefore have a factor of safety of only 1.1, which is obviously low.

15 In these calculations we have considered the elastic limit of the material rather than the ultimate strength, on account of the dynamic character of the load, each pinion tooth receiving nearly 1800 blows per mile. The greater the ratio, therefore, between the fibre stress and the elastic limit of the material (the other physical properties remaining of proper value), the greater the life, and some relief should therefore be obtained by the adoption of the special 20 deg. stub tooth which reduces the fibre stress nearly 30 per cent and increases the minimum factor of safety from 1.1 to nearly 1.6.

16 A much greater relief, however, may be looked for by the use of steel with a high elastic limit, say 90 000 lb. per square inch, which used in conjunction with a design of stub tooth increases our minimum factor of safety to 3.2 as compared with our present 1.1.

17 It will be noted that the foregoing calculations are all dependent upon the use of the Lewis formula, the fundamental assumption of which is that the teeth bear across the whole face. Consideration, however, must be given to the fact that in practice it is not possible always to maintain perfect alignment between gear and pinion on account of the necessarily rough design of the motor and its assembly on the truck. The effect of this disalignment is shown in Fig. 3, which is a photograph of a pinion which broke in service, and in the side view it is seen that the lower half of the teeth have been battered over by the loose piece jamming between the gear and pinion. It is the upper half of the teeth which is cracked, however, showing that these cracks were not caused by the jamming of the loose piece, but by the gradual hammering of the pinion against the gear in service. It will be noticed that points *a*, *b*, *c*, *d*, *e*, gave away successively until at last the loose piece was caught and ripped out, and we find that this is typical of all the failures. The break ordinarily starts at the inner face of the pinion, due to the natural tendency of the arma-

Fig. 3 16-Tooth Pinion Broken in Service

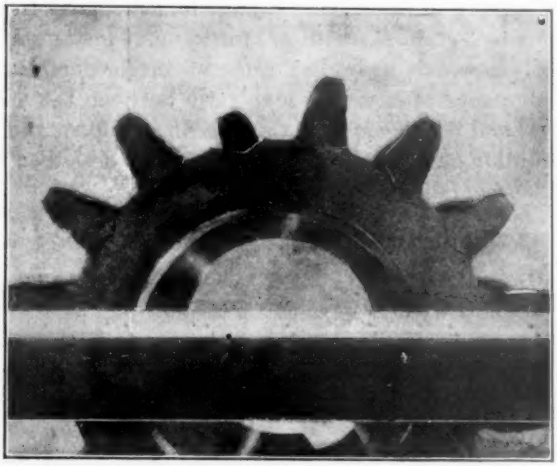

Fig. 4 End View of Pinion Shown in Fig. 3, Showing Cracked Teeth

ture to cock that way, but in some cases it starts at the outside, and is probably caused by running a new pinion with a gear which has previously worn taper.

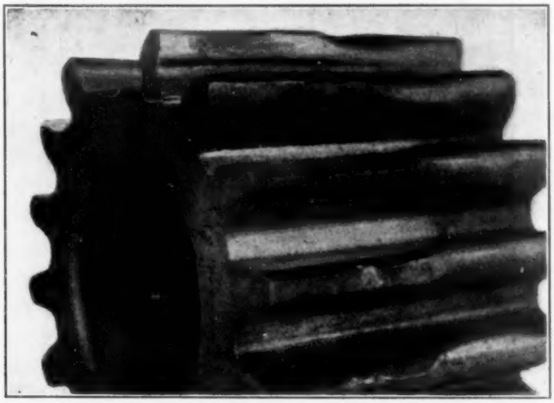

FIG. 5 THREE-QUARTER VIEW OF PINION SHOWN IN FIG. 3, SHOWING CRACKED TEETH

18 Furthermore the breakages cannot be attributed to so called crystallization or "gradual fracture," for in many instances the teeth are practically new, having broken after an inconsiderable mileage.

19 The necessity of having a high ratio between the theoretical working stress and the elastic limit thus becomes evident and justifies us in the adoption of the special means we have described to overcome the abnormal failures.

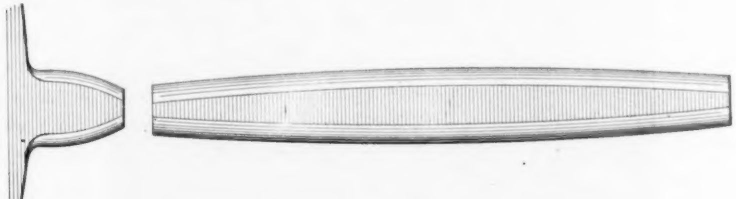

FIG. 6 SUGGESTED FORM OF TOOTH

20 The lack of alignment of gear and pinion above mentioned is probably one of the chief causes which have necessitated the use of these special means, and we have therefore endeavored to find some other method of overcoming the difficulty. This led to the suggestion

that as we are actually obtaining only a short bearing on the tooth it would be better to assure this bearing being at the center of the face rather than at the end, which might be accomplished by the use of a tooth similar to that shown in Fig. 6, in which the central portion remains the standard outline and the balance is tapered off both ways toward the ends. No investigation has been made as to the practicability of cutting such a tooth or as to its desirability, and it is suggested for consideration merely as one possible way of compensating for the lack of alignment occurring in railway motor gearing.

DISCUSSION

Mr. F. V. Henshaw Gearing for large railway motors offers problems worthy of study by specialists in steel-making and experts in gear-design. Mr. Litchfield contributes some data thereon, derived from what is, considering all conditions, probably the limit of hard service on spur gear drives. Railway motors are limited in dimensions by unchangeable conditions, the fundamentals being largely the same on great railroads and on street car lines. The available space is ample for the small motors (25 to 60 h.p.) required by the latter, gears of liberal dimensions can be provided, and the mechanical problem is confined to reducing wear: in designing motors of 150 to 200 h.p. under the same general limitations, however, the conditions are so far reversed as to require a new departure in gear practice.

2 The situation is illustrated by the following data from three typical cases in practice. The calculated values are in round numbers and based on rated loads at 500 volts. Incidentally, while horse power ratings have much the same meaning in the case of railway motors as when applied to steam boilers, yet they serve the present purpose as a fair measure of the maximum working torque.

TABLE 1 DATA OF TYPICAL RAILWAY MOTORS IN NEW YORK

Specifications	Surface Car	Manhattan		Subway
Horse power	40	125		200
R. p. m. armature	500	560		500
Torque, inch-pounds	5000	14000		25000
Torque, pounds at p.d	1770	4670		6250
Feet per minute at p.d	740	880		1045
Pitch	3	3	2.5	2.5
No. pinion teeth	17	18	16	20.
Face, gear and pinion	5	5	5	5.25
Sq. in. tooth at p.d	2.62	2.62	3.14	3.3

3 A committee of the A. S. & I. Railway Association has recommended the following gear sizes as general standards:

For motors of	45	to 100 h.p.,	3	diametral pitch,	5	in.	face
" "	"	100	" 250	"	2.5	" "	5.25	"	"

4 In short, owing to various limiting conditions, there is no kind of proportion between the motor capacities and the gear dimensions in use. The conditions for large motors are even more severe than indicated above as they actually run on higher voltages and consequently at higher speeds than stated; this fact accounts for the higher pinion velocity given by Mr. Litchfield. By the use of interpole motors with forced ventilation still greater motor capacity can be obtained with the same available motor space so that cases may arise in which gear-strength will be the governing factor in large motors.

5 As to the solution of the problem, three suggestions are quoted in Par. 10, and the author offers a further suggestion in Fig. 6 for overcoming the inevitable lack of shaft alignment. The first suggestion is the use of a coarser pitch and any possible misconception of the situation may be cleared away by the broad statement that any improvement involving reduced gear-ratios or increased gear-diameters may as well be ruled out. The highest practicable gear-ratios are none too high for many cases; there is little or nothing to spare in reducing either the number of teeth or the diameters of pinions; gear cases have no clearance to spare, and we cannot help the situation by using larger wheels. It appears then that the solution of the problem of satisfactory gearing for large railway motors must be looked for in the material and form.

Mr. John Thomson The accompanying diagram (Fig. 1) indicates a solution of the problem in Par. 20 of this paper, that can readily be accomplished by the use of ordinary gear-cutters. This consists in curving the periphery of both the pinion and its gear, each blank thus becoming a spherical zone, as *A*. Then when forming the leaves of the gear, either the blank or the cutter is to be swung in the arc of a circle corresponding to that of the peripheral surface, say as *B*, with *C* as the axis.

2 The consequence of this would be that in first starting the operation of such a pair of gears, there would be point-contact only at their centers; but as these wear away the area of contact would increase. When using involute contours, I do not see that this would affect the action detrimentally; on the contrary as new surfaces

were brought into contact the pitch-ratio would be maintained constant. Moreover, incorrect initial alignment, or that resulting from wear and tear in practice, would seem to be met automatically.

3 I have not tried this method and the idea is an off-hand shot; but if it is worth the while, anyone is welcome to use it.

MR. J. KISSICK, JR.[1] There is, to the writer's mind, a remarkable similarity between the problems of the elevated and of the subway service. The greater part of the problem will be solved for both

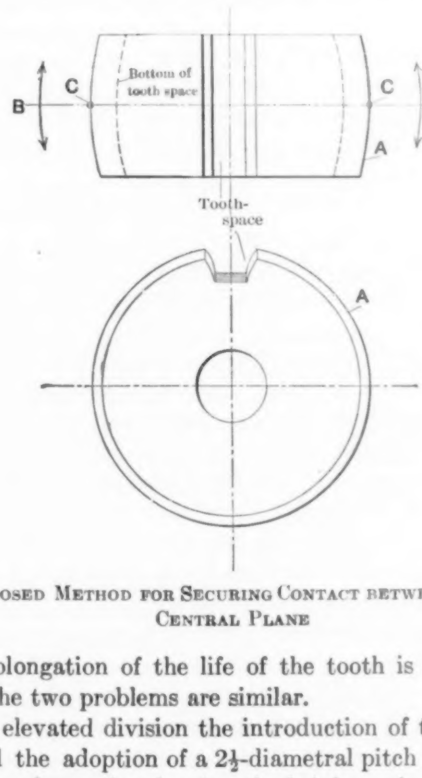

FIG. 1. PROPOSED METHOD FOR SECURING CONTACT BETWEEN TEETH AND A CENTRAL PLANE

when the prolongation of the life of the tooth is accomplished: in this respect the two problems are similar.

2 On the elevated division the introduction of the wrought steel rim gear and the adoption of a $2\frac{1}{2}$-diametral pitch have solved the question of breakage, thus leaving the brief wearing-life of the teeth as the final obstacle to be overcome.

3 Experience on the elevated road naturally caused the adoption of similar gearing equipment in the subway, although with different results. Here, we understand, the trouble seems to be breakage in

[1] With the Atha Steel Casting Co., Newark, N. J.

the pinion teeth, and to a lesser extent in the gear teeth. Since it has been deemed unsafe to run a pinion which has worn to $\frac{3}{16}$ in. at the top, it is obvious that any remedy which will prolong the wearing-life of the teeth will also prevent the large number of failures due to a high fiber stress induced in the worn tooth by the accelerating torque.

4 Mr. Litchfield has considered the problem, it seems to the writer, merely as one of a cantilever beam, the tooth being loaded at the crest and supported at the root. Being a case of simple flexure, one needs but to vary one or all of three things to enable the tooth to support a given load:

a Increase the depth of the beam, which, however, would be at the expense of the meshing tooth, although by increasing the angle of obliquity, as in the 20-deg. involute stub tooth, the tooth may be thickened to a certain extent at the root.

b Increase the elastic limit, which has been accomplished by the substitution of wrought rim gears.

c Decrease the moment by shortening the lever arm of the load, which has been brought about by the use of stub teeth.

5 Were the problem one which concerned the supporting of a static load, the above remedies would be sufficient; but being also a gearing proposition, other factors enter in that have to be met. There is no doubt that the use of steel having an elastic limit of 90 000 lb per sq. in. and the use of 20-deg. involute stub-teeth, while they have not eliminated all the trouble, have at least caused it to appear in mitigated form. The difficulty seems to be that these remedies do not go far enough, and do not treat the problem from a gearing standpoint as well as from that of a loaded beam.

6 An extended experience covering railroad work and crane service (in the latter case breakages of ordinary cast steel gears and pinions causing a menacing situation) has brought to light several factors not ordinarily considered as affecting the strength of gear teeth.

7 The first of these is resistance to wear. This, of course, needs no explanation, for it can be very easily realized that with a thick tooth there is less chance of failure than with a tooth which has been worn thin.

8 The second is permanency of form and resistance to deformation, the two being more or less related. Ordinary conditions of construction and service preclude any possibility of theoretical engage-

ments of involute teeth, and we have the substitution of sliding for rolling motion. The result, naturally, is abrasive action, which ordinary steel gears are unable to withstand; the tooth curve below the pitch line becomes flattened and slightly undercut, the amount depending upon the smallness of the pinion and consequent obliquity of action. The loss of the involute curve and the additional backlash destroy the intended action of the tooth, and instead of one set of teeth taking the load before another lays it down, we have one set carrying the whole load and transferring it bodily to the succeeding set at some place near the pitch point at the rate of some 1800 blows per minute. The effect of this hammering is commonly called "wear," but nothing could be further from the truth. No ordinary steel could stand up under these blows, and consequently we find the metal flowing towards the edges and top of the tooth depositing small slivers eventually in the gear case. An examination of the accumulation at the bottom of the tooth would show these slivers, and also a deposit somewhat similar to anvil scale.

9 The fact is that the ordinary steel gear does not wear, in the true sense of the word. Its inability to keep its shape under the severe hammering exaggerates the fault as time goes on, and premature failure is the final result. The use of stub-teeth does not help this condition any, for even the most perfectly cut tooth of this design transmits power with a very jerky motion.

10 A new steel pinion placed in service with a half-worn gear does not exert a corrective effect on the gear tooth, but instead, falls a prey to the evil tendencies of the gear. The problem therefore might be stated as follows:

- *a* A $2\frac{1}{2}$-diametral pitch 20-deg. involute stub-tooth affords a very strong tooth design.
- *b* Metal to withstand 90 000-lb. fiber stress at the elastic limit is essential.
- *c* The material comprising the tooth, in addition to sustaining the above fiber stress, (1) should be proof against wear by abrasion; (2) should offer maximum resistance to deformation, thus retaining as long as possible its original form; (3) should not be brittle, or fracture under repeated blows.

11 The latter clause really means the substitution of a special steel. Manganese steel fulfills all the requirements, being tough and extremely hard, and having an extremely high elastic limit.

12 Its wearing properties are well known, for it is the only metal

which can be used successfully in crushing-machinery, and in fact in all places where abrasion takes place. Its resistance to deformation is one of its peculiar characteristics, limiting its use in ordinary commercial work, but making it particularly valuable for the service under discussion. It has been demonstrated in comparative tests that the more prominent this characteristic is, the longer the life of the gear. Where the gear was made very soft, the size of the burr on the edge of the tooth was, to a certain extent, inversely proportional to the length of service.

13 The early considerations affecting the designs of manganese gears were naturally foundry ones, which required an even and comparatively thin metal section. Experience has overcome the casting difficulties, however, and the manufacturers have been able to turn their attention to the character of the work required of their product.

14 It is the writer's firm conviction that the use of manganese steel gears, properly designed, will solve this particular problem.

Dr. George William Sargent[1] Much stress has rightly been laid upon the desirability of stronger material that the motor equipment of the subway may obtain relief from its present gear and pinion troubles. Resistance to wear seems to me of the greatest importance in modifying the stresses to which the tooth of the gear or pinion is subject. Wear on the teeth develops play, with a consequent change in the character of the stresses. The load which was delivered statically is now delivered dynamically and must be reckoned upon as a blow. Energy expended in the form of blows tending to rupture a piece of steel is much more destructive than the same energy gradually applied. Hence the utmost resistance to wear is a desideratum as much to be sought as increased strength; and do not the nature of the breaks described in Par. 17 prove this to be the case?

2 Fig. 3 is illustrative of all the failures. "It will be noticed that points *a*, *b*, *c*, *d*, and *e* gave way successively until at last the loose piece was caught and ripped out." (Par. 17.) Such a break produces a detailed fracture not inconsistent with the statement that, as wear increased the play, thereby causing greater severity of the blows, the tooth began to rupture, ruptured in a small degree, then in a larger degree, and at last completely. To prevent absolute failure, resistance to wear is of the utmost importance, and this leads to a consideration of hardness as a quality determinative of ability to withstand abrasion or wear.

[1] With the Carpenter Steel Co., Reading, Pa.

3 It is well known that as hardness in steel increases, brittleness develops, hence there is a hardness limit beyond which, on account of the product's lack of toughness, it is not safe to go. This limit varies with each kind of steel. The carbon steels have the lowest values for hardness and toughness at this limit, thus a 45 000 lb. per sq. in. elastic limit carbon steel has a hardness of 191 (Brinell scale), a 90 000 lb. per sq. in. elastic limit carbon steel a hardness of 286, and 140 000 lb. per sq. in. elastic limit alloy steel a hardness of 400. In the first instance the values are about the maximum obtainable from an ordinary carbon steel; in the second from an extraordinary carbon steel; and in the last instance the values might be raised a little before the point of safety would be exceeded. Although it is obvious which steel is superior, consideration of the lives of the gears made from the respective steels may make it yet clearer.

4 Assuming, although it is not by any means the case, that a body twice as hard as another will resist wear twice as long, a gear made of the extraordinary steel, with its elastic limit double that of ordinary steel and its hardness increased 50 per cent, should have its life doubled on account of its increased strength and made half as long again on account of its increased resistance to wear, making a total increase in the longevity of the gear of two and one-half times. With the alloy steel gear, reasoning after the same manner, the life will be increased over that of the ordinary steel gear twice by the hardness and three times by the strength, a total of five times. These figures are too low for the facts, but they show relative values and the importance of the matter of hardness. It is stated that in practice it is impossible to maintain perfect alignment between gear and pinion. Too great hardness is therefore to be avoided, since the points of contact might be made too few, bringing excessive strain upon these few points, with the resultant rupture.

5 Failures of machines in service usually result from one or more general causes, two of which are, improper design and failure to select the proper constructive material. These are usually due to a lack of appreciation of the conditions of service, although they may be due to the development of changed conditions of service. The design may be correct and the material of construction right, and either improper methods of handling the material or imperfect manufacture of the material, or both, may yet bring about failure. Judging from the fact that the factor of safety is but 1.1, and a change in the design raises this factor 1.6, both improper design and the failure to select proper constructive material seem to be responsible for the conditions.

That this is the case is further indicated by the character of the fracture described in Par. 17 as typical of all the failures.

6 It is in just such instances that the alloy steels, with their greater possibilities, find their usefulness. Increased dimensions are not permitted, altered designs effect but a partial relief, and a harder and stronger constructive material must be used. True, these alloy steels are more expensive in their first cost, but really less expensive in the final reckoning. Where life hangs in the balance, cost should be not even a secondary consideration.

Prof. F. De R. Furman Mr. Litchfield's suggestion of giving the tooth a double-curve form can hardly solve his problem, for it will still leave the entire load of the teeth concentrated on point contact under all conditions. If the contact point should move to one side, due to improper alignment of the wheel shafts, the pressure would come at a reduced and therefore weakened section.

2 Using the data given in this paper, I have laid out the proportions of teeth obtained by standard rotary cutters (according to proportions given in Kent) for both 14½-deg. and 20-deg. pressure angles, which the author states were used, and then compared these teeth with others of special form which figure out to be much stronger. The comparative forms are shown in Fig. 1 and 2, in which teeth of standard proportions are shown entirely in section, the proposed form with shaded edges. In Fig. 1 the 14½-deg. pressure angle is used, and in Fig. 2 the 20-deg. angle.

3 I have made the comparison by considering the tooth as a beam fixed at one end, and have drawn within each tooth a parabola having its vertex at the middle of the crest and its two branches tangent to the outline at or near the root. Then applying the formula $hP = \frac{1}{6} Fbt^2$, in which h = height of tooth, P = pressure on tooth at the tip, F = working stress in tooth, b = breadth of tooth (taken as 3 in.), and t = thickness of tooth, we have for F = 39 400 lb. (the figure given by Mr. Litchfield), P = 7700 lb. for the standard tooth having a 14½-deg. angle. For the proposed tooth with the same angle, taking P = 7700 as above, we find F, the stress at the root = 23 600 lb., against 39 400 for the standard tooth, an increase in strength of 67 per cent in favor of the special form of tooth.

4 For the tooth with a 20-deg. angle we find for the same value of P (7700 lb.) that the stress in the standard tooth is 35 600 lb., against 25 600 lb. for the proposed tooth, an increase of 39 per cent. While the 20-deg. tooth shows up the stronger for the standard form, the

14½-deg. tooth is the stronger in the proposed form. This is due principally to the decreased length of tooth for a given path of action, made possible by the 14½-deg. angle, and also to the specially formed fillet which starts from a point well up from the root circle.

5 The amount of action in the 14½-deg. proposed tooth is about 1.5, practically the same as for the standard form, assuming that that part of the standard tooth which is above the interference line is not in action.

6 The large fillets at the roots of the proposed teeth were found by

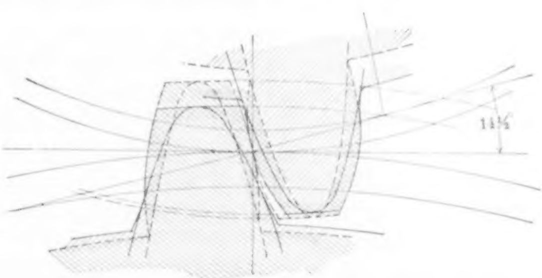

Fig. 1 Comparison of Tooth Forms 14½-Deg. Angle

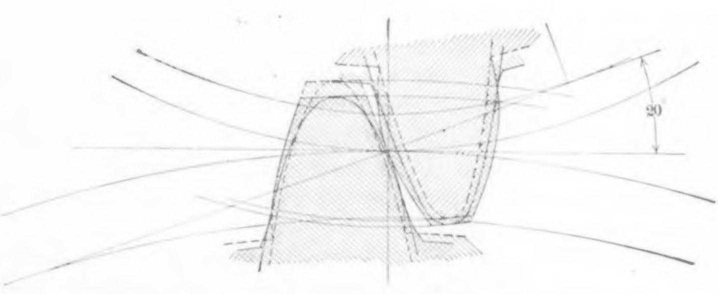

Fig. 2 Comparison of Tooth Forms 20-Deg. Angle

tracing the path of the corner of one tooth on the plane of the other wheel and then placing the tooth outline within this path. The parabola was then drawn tangent to the tooth outline. In making the above computations the weaker tooth on the smaller wheel was used in each case.

7 It will be noticed that the addenda for the two teeth are not equal, a point referred to in my discussion of Mr. Flanders' paper.

THE AUTHOR It was not the purpose of the writer to attempt to approach this subject from the standpoint of the gearing-expert but merely from that of the user of a considerable amount of gearing material on an exceedingly intense service, and to lay before the Society those encountered difficulties which might be of interest, together with the efforts made to overcome them.

2 While it is impossible to take up the discussion in detail, in the limited time allowed for review of the various points, correction should be made of a seeming misapprehension on the part of one writer who states: "There is no doubt that the use of steel having an elastic limit of 90 000 lb. per square inch and the use of 20-deg. involute stub-teeth, while they have not eliminated all the trouble, have at least caused it to appear in mitigated form."

3 At the time of the preparation of the paper practically none of the gearing under our observation was of the stub-tooth, high elastic-limit type, and while since then a large number have been installed, the length of service has obviously been too short for judgment of the value of the improvement.

4 Much stress is laid by others on the value of alloy-steels, and it should be stated therefore that the attractiveness of several of them has been such as to lead to a thorough investigation of their merits, which is still under way. These investigations include actual service tests in quantities large enough to give a fair indication of results to be expected from a general adoption of the particular class of steel in question. Until these data, coupled with those yet to be found for the treated carbon-steel, are obtained, it would be improper to discard the carbon-steel and adopt the alloy.

5 Aside from the question of the tooth form, the situation may be thus briefly summarized:

The carbon-steel has failed and data must be obtained from service of the resistance to failure of *a* specially treated carbon-steel; *b* alloy steel.

If one shows a manifest superiority over the other, then that should be adopted, but if they compare favorably, the problem will then have changed from one of failure to one of wear, and we will at the same time be in possession of the information necessary to determine this second phase of the question of gearing on heavy railway equipments.

No. 1220

ARTICULATED COMPOUND LOCOMOTIVES

By C. J. Mellin, Schenectady, N. Y.

Member of the Society

The constantly increasing demand for heavier power, made by most railways in the country during the last decade, and especially by those roads having heavy gradients combined with sharp curves, brought out various designs which on account of rail pressure limitations required so many coupled wheels that the length of the rigid wheel base made them unwieldy to operate with efficiency. This demand for greater power was, of course, greatest in mountain districts where heavy grades and sharp curvatures generally go together, necessitating, for safe operation, comparatively short wheel bases, reduction in engine resistance and wear of wheel flanges and rail, together with moderate weight of the working parts of the engine.

2 In striving to meet this demand, the locomotive designers and builders were brought face to face with an insurmountable barrier to further progress in the enlargement of engines on the old lines; and in 1902 the American Locomotive Company decided to work out a design of a heavy, powerful locomotive for the Baltimore and Ohio Railroad, having two sets of engines under one boiler, capable of adjusting themselves independently to the alignment of roads with curvatures up to 30 deg. on the principle developed by the prominent French engineer, M. Anatole Mallet of Paris.

3 Mr. Loree, then President of the Baltimore and Ohio Railroad, considered the question seriously; but it was first thought that it would be of no advantage to the Baltimore and Ohio Railroad, even if it proved successful, and the subject was left undecided for some time. In the latter part of 1903, on the recommendation of Mr. J. E. Muhlfeld, who in the meantime had become General Superintendent of Motive Power, the Baltimore and Ohio ordered one engine of this type, which was built at the Schenectady Works of the American Locomotive

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

Company during the winter of 1903 and 1904, to suit the conditions of that railway.

4 Propositions for building the Mallet type of engine had been made both by S. M. Vauclain, General Manager of the Baldwin Locomotive Works and by the writer, several years prior to the above date, but the one under consideration is, so far as the writer is aware, the first engine of this type completely designed and constructed in the United States. The original Mallet type of engine built in Europe dates back to the early nineties, but its history would be too extended to embody in this paper.

5 This Baltimore and Ohio locomotive, which was of unusual dimensions for that time, was exhibited at the Louisiana Purchase Exposition at St. Louis in 1904. A great number of locomotives of this type, of various sizes, have since been built for other roads as forerunners of what promises to be the most powerful and efficient type of the freight engine of the future. The form permits of the application of cylinders of largest dimensions, as well as of the largest boiler capacity, by the distribution of the weight over a long wheel base and over many driving axles. Up to the present date a tractive power of about 125 000 lb. has been reached, whereas the ordinary types of engines rarely exceed 44 000 lb.

6 The tractive power P is figured by the usual formula for multiple expansion engines working compound, namely:

$$P = \frac{d_L\, p\, c\, s}{D}$$

in which

d_L = Diameter low pressure cylinder.
p = Boiler pressure.
s = Stroke of piston.
D = Diameter driving wheels.
c = Coefficient obtained from actual practice, and varying with the degree of cut-off and cylinder ratio.

The cylinder ratio being as a rule 1 to $2\frac{1}{2}$, and the cut-off in the high pressure cylinder about 82 per cent the value of coefficient c is about 0.53 at a piston speed not exceeding 250 ft. per minute.

7 For working single expansion, the formula will be that of a simple engine multiplied by 2 or

$$P^1 = \frac{2\, d^2\, p\, c\, s}{D}$$

where the symbols are the same, except d, which represents the diam-

eter of the high pressure cylinder, and the value of c equals 0.85, which is accepted for simple engines.

8 The factor 2 is used on the basis that the low pressure engine develops the same power as the high pressure, because the steam supply to the former is reduced in an inverse proportion to the cylinder ratio.

TABLE 1 COMPARISON OF PRINCIPAL DIMENSIONS AND WEIGHTS OF INDIVIDUAL PARTS OF HEAVIEST MALLET ENGINE AND HEAVIEST DESIGNS OF ORDINARY TYPES

Road	Erie	D. & H.	P.B. & L.E.	A. T. & S. F.	A. T. & S. F.	B. R. & P.
Builder	A. L. Co.	A. L. Co.	A. L. Co.	A. L. Co.	Baldwin	A. L. Co.
Type	Articulated 0880	Consol. 280	Consol. 280	Decapod 2100	Decapod 2100	Decapod 2100
Simple or compound cylinders	Comp. 4 cyl. articulated	Simple	Simple	Comp. 4 cyl. tandem	Comp. 4 cyl. tandem	Simple
Diameter and stroke	25 & 39 by 28 in.	23 by 30 in.	24 by 32 in.	17½ & 30 by 34 in.	19 & 32 by 32 in.	24 by 28 in.
Boiler pressure	215 lb.	210 lb.	210 lb.	225 lb.	210 lb.	210 lb.
Diameter of driving wheels	51 in.	57 in.	54 in.	57 in.	57 in.	52 in.
Tractive power	94 800 lb.	49 690 lb.	60 900 lb	55 300 lb.	62 500 lb.*	55 350 lb.
T. P. working simple	120 000 lb.			63 000 lb.	64 100 lb.	
Total weight	410 000 lb.	246 500 lb.	250 500 lb.	259 800 lb.	266 500 lb.	268 000 lb.
Weight on drivers	410 000 lb.	217 500 lb.	225 500 lb.	232 000 lb.	237 000 lb.	243 000 lb.
Factor of adhesion	4.33	4.38	3.7	4.20	3.82	4.39
Rigid wheel base	14 ft. 3 in.	17 ft.	15 ft. 7 in.	20 ft.	20 ft. 4 in.	19 ft.
Average load per wheel on rail	25 625 lb.	27 200 lb.	28 200 lb.	23 200 lb.	23 700 lb.	24 300 lb.
Weight of main rod	968 lb.	985 lb.	1144 lb.	817 lb.	1028 lb.	1050 lb.
Weight of inter. rod	406 lb.	681 lb.	424 lb.	574 lb.	x	449 lb.
Weight of front rod	133 lb.	188 lb.	260 lb.	158 lb.	x	187 lb.
Weight of back rod	135 lb.	208 lb.	200 lb.	158 lb.	x	190 lb.
Weight of back inter. rod				244 lb.	x	190 lb.
Weight of h.p. piston and rod	790 lb.	782 lb.	693 lb.	1130 lb.	1075 lb.	830 lb.
Weight of l.p. piston and rod	993 lb.					

* Calculated by the formula used by the Baldwin Locomotive Works for figuring the tractive power of 4-cylinder tandem compound locomotives.

9 The Mallet articulated arrangement presents the advantages of enormous tractive power concentrated in the combination of the two sets of engines with practically no increase in the individual weights of the moving and wearing parts over those of engines of the ordinary

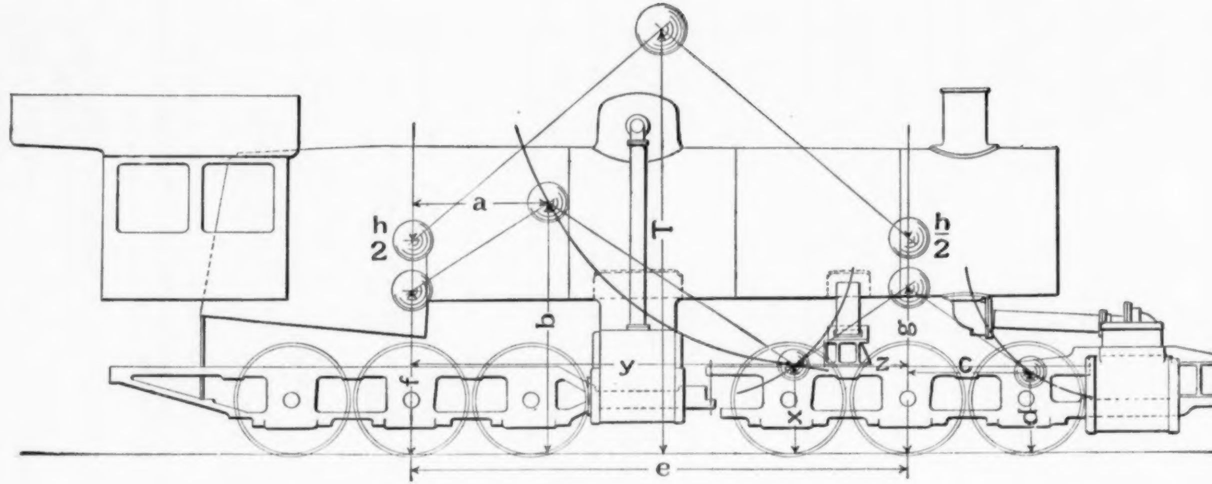

FIG. 1 WEIGHT DISTRIBUTION OF ARTICULATED LOCOMOTIVE

$a \times b = y \times x \qquad c \times d = z \times x \qquad (a \times b) + (c \times d) = e \times x$

$$x = \frac{(a \times b) + (c \times d)}{e}$$

$f = b - x \quad f = g \quad g = d \quad g = d + x \quad h =$ d ad weights

$$f + \frac{h}{2} + g + \frac{h}{2} = T$$

types; double expansion of the steam; simplicity and ease in operation; and a short rigid wheel base, with the weight distributed over a long total wheel base, resulting in the greatest flexibility and ease on track and bridges. It was also found possible at the very first to provide an engine under the control and operation of a single crew, having double the power of the largest engines of the ordinary type.

10 These advantages are most clearly evidenced by the comparisons in Table 1 between the heaviest engine of the Mallet type ever built and some of the heaviest freight engines of standard types.

11 The comparison in Table 2, however, between one of the lighter designs of Mallet engines and a few engines of the ordinary types, of approximately the same power, clearly shows that the same tractive power is obtained with this type as in other types, but with weights of moving and wearing parts equivalent to those of an engine of half the tractive power.

12 In general, the Mallet articulated locomotive consists of a front system with its frame work, low-pressure cylinders with their

TABLE 2 COMPARISON OF PRINCIPAL DIMENSIONS AND WEIGHTS OF INDIVIDUAL PARTS OF MALLET ENGINE AND DESIGNS OF ORDINARY TYPES OF FREIGHT ENGINES HAVING APPROXIMATELY SIMILAR TRACTIVE POWER

Road	Central Railway of Brazil	N. Y. C. Lines	Erie
Builder	A. L. Co.	A. L. Co.	A. L. Co.
Type	Articulated 0660	Consolidation 280	Consolidation 280
Simple or compound	Compound 4 cyl. articulated	Simple	Simple
Cylinder diameter and stroke	17½ in. 28 by 26 in.	23 by 32 in.	22 by 32 in.
Boiler pressure	200 lb.	200 lb.	200 lb.
Diameter of driving wheels	50 in.	63 in.	62 in.
Tractive power	42 420 lb.	45 700 lb.	42 500 lb.
Tractive power working simple	52 000 lb.		
Weight on drivers	206 000 lb.	208 000 lb.	179 000 lb.
Total weight	206 000 lb.	234 000 lb.	202 000 lb.
Factor of adhesion	4.85	4.5	4.2
Rigid wheel base	9 ft.	17 ft. 6 in.	17 ft.
Average load per wheel on rail	17 166 lb.	26 000 lb.	22 375 lb.
Weight of main rod	417 lb.	821 lb.	848 lb.
Weight of intermediate rod		403 lb.	509 lb.
Weight of front rod	208 lb.	185 lb.	246 lb.
Weight of back rod	92 lb.	201 lb.	210 lb.
Weight of h.p. piston and rod	297 lb.	664 lb.	516 lb.
Weight of l.p. piston and rod	459 lb.		

attachment of guides and rods, one set of driving wheels, cross-ties and boiler supports; and a rear system with the high pressure cylinders and frame work, fixed to the boiler as though an ordinary engine were placed under the rear part of the boiler.

13 The supporting points are of course in the center of their respective wheel bases; and are preferably used as the reference centers for calculating the moments of the individual systems.

14 It will be seen from Fig. 1 that, when the proper weights have been arrived at as a whole, the weight of each system with its center of gravity must be figured out independently, starting with the front system in which the center of gravity falls well ahead of its center of support. As the rear system is by far the heavier, the front system must therefore carry one half of the difference between the two to divide the weight equally. The point where this weight is to be supported on the front system is readily located by finding the required lever arm to obtain the same moments in the rear of the center of the wheel base as those in front of it. This center may be called the virtual supporting point of the rear system on the front engine.

15 The weight on this point, multiplied by its distance from the center of the wheel base of the rear system, gives the moment for this system. By dividing the weight of the rear system into this moment, we find the distance from the wheel base center of that system at which its center of gravity must be located; as is graphically shown, with all moment curves in both systems, in Fig. 1. The balls in the figure represent the various weights so far as they effect the distribution, and their relative value is represented by their height from the base line. It is not necessary to include wheels, axles, and boxes or what are termed dead weight in this method of calculation, because they are all central over the supporting points; and, as shown in the figure, are added after the distribution of the live weight is ascertained. In Fig. 1, the curve to the right of the front wheel base center is the moment curve of that system. This curve is reproduced on the left of the center and will pass through the point where one-half of the distance between the two systems will balance that of the front system. Starting from this point, and with reference to the rear wheel base center, we get the moment curve for the rear system which passes through the center of gravity of that system where it intersects the level from the common base line, representing its total weight. All the moment curves are of course true hyperbolae and are therefore easy to construct. An approximate plan is laid out and modified until the desired total weight is obtained; and,

if the center of gravity of the rear system does not coincide with this calculation of distribution, it is generally most convenient to shift the location of the boiler in the required direction until that occurs; or, if preferred, the framing and heavier castings in front and rear of the center of gravity may be modified to produce the same effect.

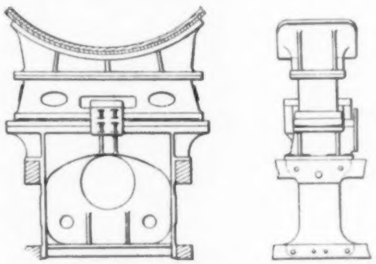

Fig. 2 Main Boiler Bearing

16 The support of the rear system on the front engine, shown in Fig. 2, must be capable of sliding laterally to allow the front engine to enter a curve and form an angle with the rear engine. For this reason the two engines are hinged together by vertical swivel pins located, preferably, slightly ahead of a point midway between the two wheel sets, because the swivel point being so located aids the

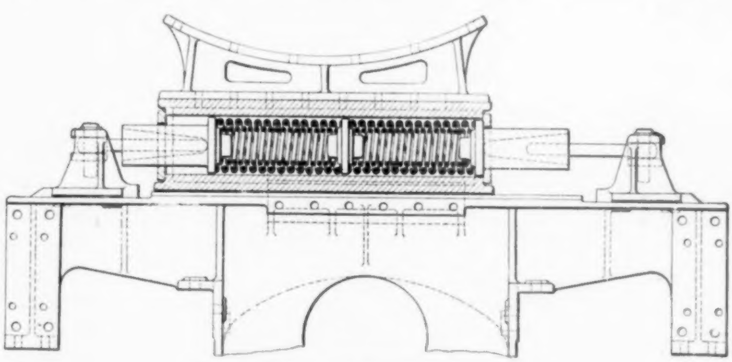

Fig. 3 Spring Centering Device

guiding of the rear engine into curves on account of its closer proximity to the center line of the track when the engine is going ahead. A lateral soring applied in the front part of the leading engine permits an elastic yielding of that engine into a curve and restores alignment in leaving it (Fig. 3).

17 It is not practical, however, to place the sliding support on the virtual location previously mentioned; because the front engine would then be unstable and tip one way or the other, like a scale beam, with the slightest disturbance due to change of weight on grades, but it should be on a convenient location in front of the virtual one. This, of course, would allow the front engine to tip forward as the imposed weight on that point would be less and the arm from the driving wheel base center, shorter. To correct this disturbance, a pair of vertical hanger bolts are applied between the upper member of the frame at the extreme rear end of the front engine and the lower member at the extreme front end of the rear frame (Fig. 4); or two supports may be provided, one on each side of the virtual supporting point. With these bolts or hangers, the

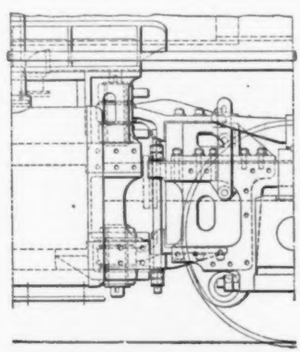

Fig. 4 Vertical Hanger Bolts

proper alignment of the front engine may be adjusted at will and the effect of the combined supports is restored to the virtual supporting point. This method reduces the pressure on the sliding plate and imposes load on the bolts in proportion to the respective distances of the two supports from the virtual supporting center.

18 It is evident that the supporting bearing must be self adjusting to bear evenly at all alignments of the engine. To accomplish this, the lower, or what we may call the pillow plate, has a ground surface on its upper side on which a corresponding upper plate can slide with the movement of the boiler; while its underside is slightly curved to allow a longitudinal rocking. This pillow plate is held in position by means of two dowel pins of about 2 in. in diameter recessed about ¾ in. into the plate and an equal amount down into the cross-tie on which it rests approximately in the middle of

the convex or curved part. In this way the pillow plate is kept in position without destroying its ability to adjust itself so as to present a uniform bearing for the upper or sliding plate at any alignment of the engine, as shown in Fig. 2.

19 The front engine thus becomes a very efficient leading truck for the rear engine. Opinions on the use of a truck in the articulated engine are, however, divided; but, because of the many objections connected with the application of a front truck in freight service as to the first cost, maintenance, dead weight and unfavorable distribution of the machinery sometimes causing serious obstructions, nothing is gained by this objectionable feature; as it is practically the same as putting a truck ahead of a truck.

20 The front engine in going ahead being a truck in itself, the first pair of drivers have a leverage in their favor in entering the curve. The reason for this is that the virtual support of the weight of the rear system, which is carried by the front, falls back on the latter and in the rear of the sliding bearing; thus allowing a great part of the load of the rear engine to be carried by the hanger bolts (Fig. 4) between the two frames.

21 This alone reduces the pressure very materially on the sliding plate, which together with the short arm for friction resistance and long guiding arm for the flanges, reduces the pressure on them to a small fraction of the total friction load on the sliding plate; and comparatively light centering springs will therefore suffice for this purpose and still further reduce the flange pressure.

22 These same leverages and resistances act equally favorably in backing; as it is simply a reversed operation and the rear drivers have to swing the boiler against these resistances. Therefore, it is important that these should be small and with the shortest possible leverages, which naturally also minimizes the flange pressure on the rear wheel; that is, the last wheel of the engine, which then has to do the guiding.

23 With the use of a front truck, the center of support is shifted forward and with it the virtual and actual supporting points of the weight of the rear engine carried on the front system. The weight on this support must, therefore, be increased with the carrying capacity of the truck and offer little or no opportunity for transferring any of this load to the hanger bolts, practically doubling both the load on the sliding plate and the length of the resistance arm. At the same time, by the application of a front truck, the guiding point is moved forward so that the leverage has been increased to offset

the increased side resistance of the engine. The guiding power of the truck, however, is limited to its swing resistance. This, therefore, may leave as much or more guiding to be done by the front drivers as where no truck is used because of the increased moments of resistance of the engine in curving.

24 A more serious matter, however, is the backing with a front truck. The high resistance moments in the front must be overcome by the rear drivers, which are doing the guiding, and it is easy to understand how fast the flange pressure is multiplied by this displacement of the load and the safety margin for derailing dangerously reduced. It is therefore evident that a rear truck is a necessity when a front truck is used where backing is to be considered; thus curing one evil with another. Even with the application of a rear truck, the objections caused by the application of the front truck will be only partly compensated for; as the following very essential objections still remain:

a The application of a front truck increases the distance of the front buffers from the first pair of drivers by 15 to 20 per cent, and consequently throws the front drawhead of the engine further out from the center of the track incurves than with the shorter extensions where no front truck is used.

b It increases the total wheel base of the engine about 8 ft. 6 in., requiring an 80-ft. turn table to take an average sized engine with its tender.

c Additional dead weight to be carried by the truck must be provided and the expenses in maintenance and first cost by the use of it are items that should not be overlooked.

d The long arms for friction resistances on the sliding plate with increased load on them, due to the front truck, will not be lessened by the application of a rear truck.

e When only a front truck is applied, the boiler is necessarily moved so far forward that it leaves scant room for the valve motion on the rear engine. The result of this is that the width of the firebox is necessarily limited to about 72 in.

25 In the case of passenger engines of the articulated type, however, large wheels would be used, and only four pairs of drivers can or need be applied. A four wheel front truck, with rigid center pin and rigid trailing wheels works, in conveniently in the place of a third

pair of drivers in each engine front and rear, respectively; which otherwise with their large diameter, make the engine unduly long.

26 Continuing with the construction of an engine without a truck, the spring rigging and equalization are next to be taken into consideration. The springs are most conveniently located above the boxes and frame, except under the firebox where they must be placed between the adjacent wheels with yokes over the boxes in the usual manner. On the front engine, all springs on each side are equalized together with a cross equalizer between the front springs. The rear engine is equalized in the same manner, except that the cross equalizer is omitted. This makes a three-point suspension of the whole engine and prevents any excessive local stresses of a diagonal nature on an uneven road; as the front engine accommodates itself very

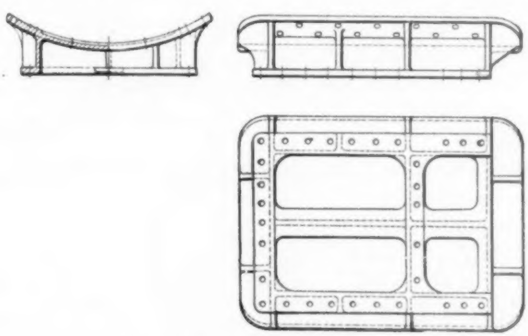

FIG. 5 BOILER SADDLE

freely to the rear engine and approximately divides the angularity between the inclination of the axles. The wheels then follow the rail comparatively freely and easily on the twisting parts; at the rising of the outer rail; on entering and leaving curves; as well as on any other unevenness of the road.

27 The high pressure cylinders, as previously stated, are located under the cylindrical part of the boiler, generally slightly in the rear of the middle of the barrel. They are provided with a cast steel saddle, bolted or riveted to the shell of the boiler by its upper flange and to the cylinder half saddle by its lower flange as shown in Fig. 5. The cylinder saddles, however, are not divided in the center of the engine; but sufficiently to one side to allow the receiver pipe with its ball joint to be placed in the center line of the engine (Fig. 6).

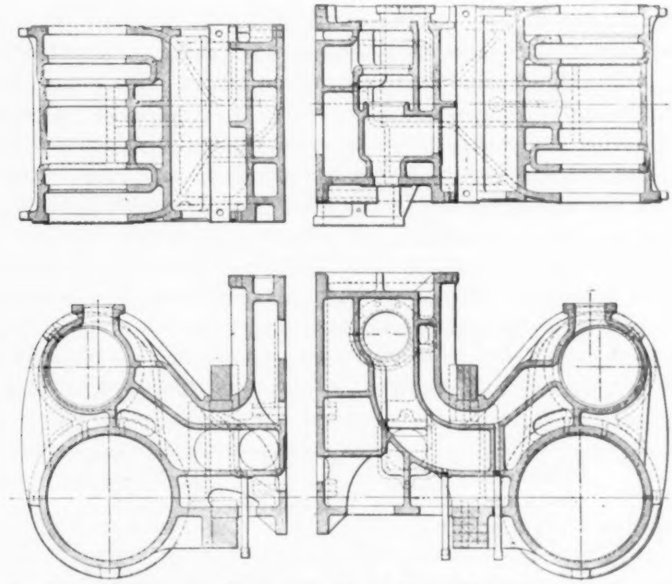

FIG. 6 CROSS SECTIONS OF RIGHT AND LEFT HAND HIGH PRESSURE CYLINDERS

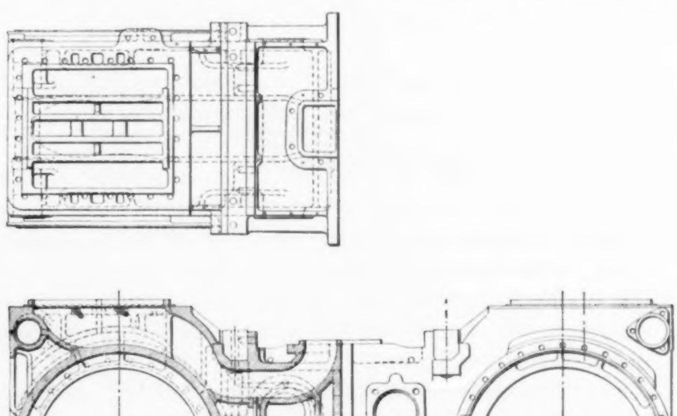

FIG. 7 CROSS SECTION, FRONT AND PLAN VIEWS OF LOW PRESSURE CYLINDERS

28 The low pressure cylinders, Fig. 7, are fixed to the front engine with so-called "half saddles" but not connected to the boiler; except by the flexible exhaust pipe. With the large low pressure cylinders required for the desired power, there will naturally be con-

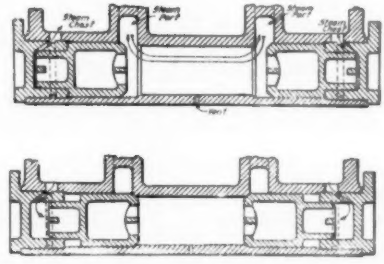

FIG. 8 BY-PASS VALVES

siderable suction of cold air through the cylinders in running with the steam shut off; which is very objectionable for several reasons, one of which is the sudden cooling of the cylinder walls and passages.

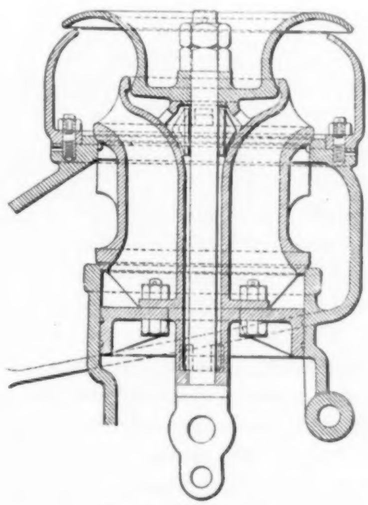

FIG. 9 THROTTLE

To obviate this, a pair of by-pass valves (Fig. 8) are located in the cylinder castings. These valves automatically open communication between both ends of the cylinder when the throttle is closed, so

that a certain amount of air can circulate through this passage to and fro with the movement of the piston, and thereby prevent the sudden changes in the temperature and relieve the pumping of air through the exhaust nozzle, and also minimize unnecessary agitation of the fire.

29 The steam enters the high pressure cylinders directly from the dome, where the throttle is located, as usual, and it exhausts into the high pressure cylinder saddles on each side and meets at a point where the steam enters the intercepting valve. After passing through into the receiver pipe placed in the center of the front engine at a

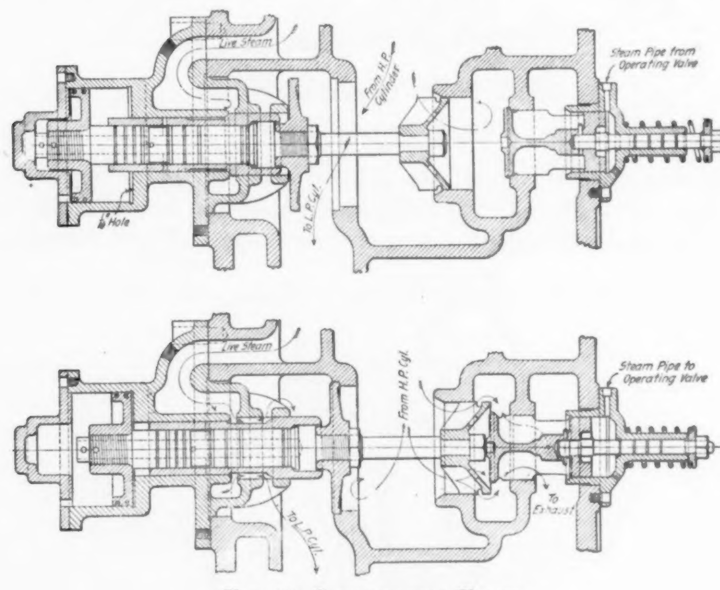

FIG. 10 INTERCEPTING VALVE

convenient height above the axles, it branches off through a "Y" pipe to each of the low pressure cylinder saddles and through passages in these saddles to the steam chest. The exhaust from the low pressure cylinders returns through the saddles and meets in a pipe delivering to the common exhaust pipe and the stack.

30 In order to obtain high efficiency and emergency power, various special details that may be of interest have been brought into use in these designs.

31 The throttle shown in Fig. 9 is provided with a steam separator at the extreme top, where the steam enters in an upward direc-

tion; and, after entering, meets a sharp turn downward whereby the entrained water is thrown against the curved walls of the crown, is entrapped at its base and forced down through a central passage back to the boiler by the inertia exerted in the trap. The continuous current of moisture that is abruptly arrested allows no chance for the water particles, once brought into contact with the curved wall of the crown, to escape the trap. The steam in relieving itself from the moisture makes the turn into the valve opening directly at the top and through the valve body to the lower opening as will be seen from the cut.

32 Following the course of the steam, we do not meet any unusual construction of the passages until we reach the intercepting valve which, although not specially designed for this engine, should be considered in this connection as a most important factor; as it controls the pressure for the receiver and low pressure cylinders, supplies steam direct from the boiler at a proportional pressure to the large cylinders in starting, prevents this pressure backing up against the high pressure piston and makes it possible to increase the

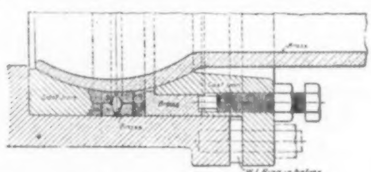

Fig. 11 Receiver Pipe Ball Joint

power of the engine about 20 per cent at critical moments, by permitting the engine to be changed in such emergencies so as to work live steam in all cylinders with equivalent distribution of the work. With the exception of changing the engine into simple while under way, all its movements are automatic (Fig. 10).

33 The intercepting valve consists principally of two valves intimately combined; the one effecting the closing of the other; and a change valve by which the former is unbalanced in turning the engine into simple working when under way. The main valve closes the receiver and prevents the reduced live steam pressure from backing up against the high pressure piston in starting and working simple, and by closing the exhaust valve the accumulation of exhaust from the high pressure cylinders automatically opens the main valve to the low pressure side of the receivers and simultaneously closes the admission and reducing valve, whereby the engine is changed into compound.

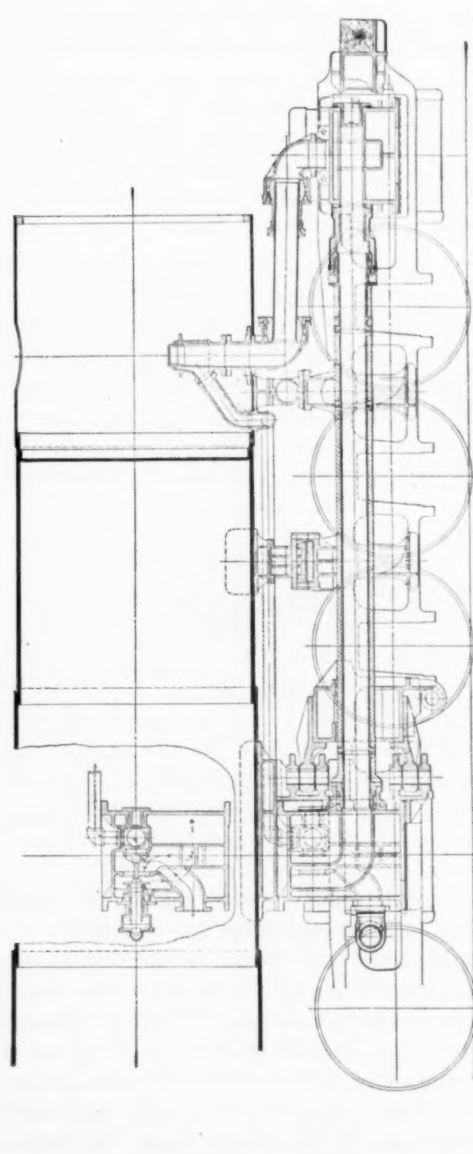

FIG. 12 CENTRAL LONGITUDINAL SECTION FROM HIGH PRESSURE CYLINDER TO LOW PRESSURE CYLINDER, SHOWING STEAM PIPES

34 The live steam admission and reducing valve has the form of a sleeve placed on the stem of the main valve; and, as seen in the cut, is allowed a limited longitudinal play to perform its double function.

35 The third, or change valve, has two functions to perform and is operated by the engineman in emergencies; and is, therefore, known as the emergency valve. Its use is resorted to only when the engine is about to stall on a heavy grade or at a difficult starting place. The first function of this valve is to unbalance the intercepting and reducing valves so that the former cuts off the low pressure side of the receiver; and the second function is that of an outlet valve for high pressure cylinder exhaust steam in working simple, which later is led in an independent pipe to the stack.

36 The next features of importance are the ball joints in the receiver and exhaust pipes which are peculiar to this type of engine. They consist (Fig. 11) of ball bearings, gland and packing. The latter is made of eight ½ in. square rings of Vulcabeston or fibrous packings laid in pairs, in the middle of which a brass ring of an elongated diamond section is inserted. Being just the width of the packing space, this ring seals all joints in the packing rings proper and forces them tightly against the inside of the box and against the ball. The receiver pipe, with the ball joint and its location, and the exhaust pipe are shown in Fig. 12.

37 The flexible exhaust pipe has two ball joints and one slip joint; as it is subject to a greater angle of deflection and elongation than the receiver pipe, which has its ball joint in the vertical center line of the pivot pin between the two sets of engines.

38 The valve motion in each set of engines is of the Walschaert type, driven by their respective main axles and crossheads and operated with a common reversing gear which simultaneously changes the motions of all the valves. Because of the lateral motion of the front engine in curving, means must be provided for flexibility in the operating gear so that this movement does not interfere with the motion of the valves. This is accomplished by using an exceptionally long lifting link, shown in Fig. 13, having a double jaw in its upper end and a universal joint or ball bearing at the radius bar, which allows its lower end to follow the movement of the engine transversely relative to the rear engine, as well as the longitudinal movement of the valve in any angularity of the engine within the required limit of the swing.

39 The valve gear is operated by a hydro-pneumatic reversing

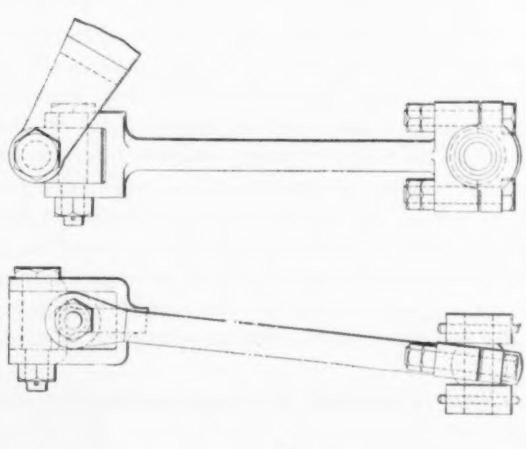

Fig. 13 Radius Bar Lifter

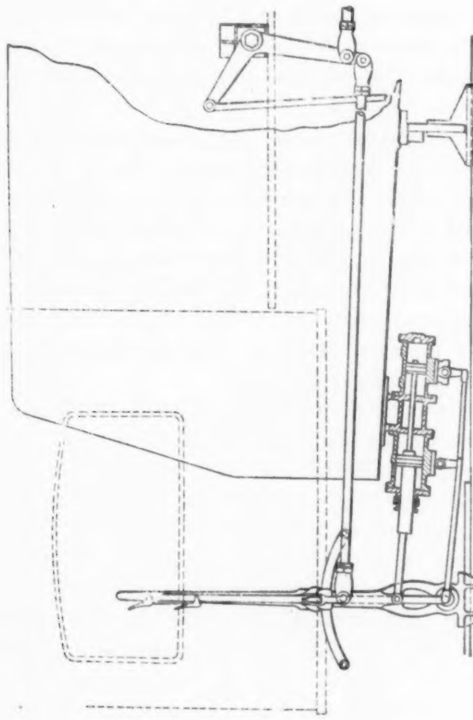

Fig. 14 Reverse Lever Arrangement

mechanism, Fig. 14 and 15, consisting of one air and one oil cylinder, with the common piston rod connected to the main reversing lever. On a suitable location on the *main* lever is pivoted a second lever for operating the gear.

40 A forward movement of this lever throws its lower end backward, turning the valves of the air and oil cylinder (shown in Fig. 15), thus making communication with the rear end of the air cylinder for air pressure to force the piston forward and, with it, the entire gear. The oil cylinder serving as a lock and regulator has, by this movement, established communication between both sides of its piston, allowing the latter to follow the movement of the gear to which it gives a moderate and uniform motion because of the contracted passage for the oil through the valve. By stopping the

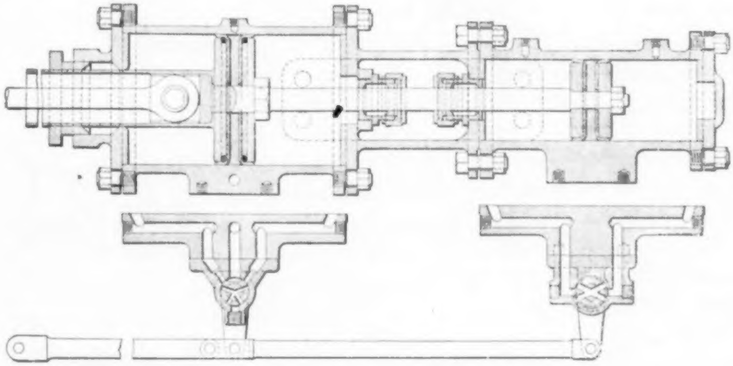

Fig. 15 Reversing Cylinders and Valves

movement of the operating lever, the gear moves the main lever up to the given relation to the former; and then, automatically, shuts off the air supply and locks the oil cylinders.

41 In unlatching the operating lever, the same movement raises the main latch which cannot drop until again in the given relation to the former, when the valves of both air and oil cylinders are closed and a positive locking of the gear is secured in addition to that of the oil lock. The handle part of the main lever is made for the purpose of operating the engine by hand in the absence of air pressure or in case of any disorder of the gear.

42 These illustrations show only a few parts peculiar to this special type of locomotive as ordinarily constructed by the American

Locomotive Company. These locomotives have proved very satisfactory in every respect, and have practically become a standard with these builders, but are of course subject to variations to suit various service conditions.

43 The most striking variation in any of these details is probably the intermittent draft coupling of this type of locomotive turned out by the Baldwin Locomotive Works, shown in Fig. 16, where a lateral play is provided for, and the draft pin is held in a central position by a series of springs. With the exception of some articulated engines built by them for service in Porto Rico in 1904, the first of this type

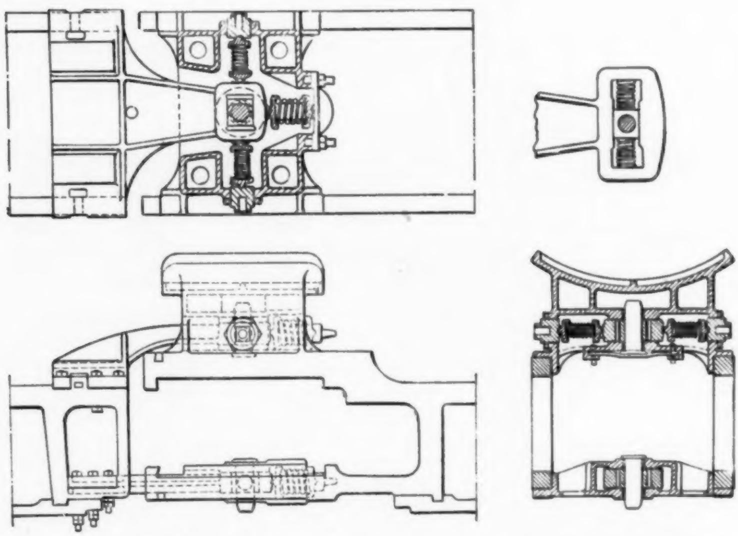

Fig. 16 Baldwin Frame Pivot Connection for Articulated Locomotives

built by the Baldwin Locomotive Works were those constructed for the Great Northern Railway; the pivot connection of which is shown in the previous figure. Of this design 67 engines are now in service on the Great Northern, 3 on the Chicago, Burlington and Quincy and 16 on the Northern Pacific. The general design is shown in Fig. 17. These engines are giving universal satisfaction and the present number is the result of repeated orders without any change whatever in design.

44 The above mentioned builders have also excluded the use of by-pass valves, double ported slide valves, and the intercepting valve

in their cylinder construction, the starting being effected by simply letting live steam into the receiver.

45 The question of superheating the steam in connection with this type of locomotive has been considered practically from the outset. Superheating would further increase the efficiency of the engine, but it has so far been deemed advisable to leave it off to avoid complications until this type becomes more generally known.

46 Fig. 18 illustrates a design of Mallet locomotive prepared in accordance with instructions from Mr. Kendrick, Vice-President of the Atchison, Topeka and Santa Fé Railroad, now being constructed under his patents by the Baldwin Locomotive Works. In this design the combustion chamber which is also fitted with superheater device is used.

47 Among the various differences between this class of engines and those of the ordinary type, is the action of this engine when loaded to the slipping point. While the former are less liable to slip than the latter, due to a more uniform pressure on the pistons, they will not be considered loaded to anywhere near their capacity until slipping takes place, and consequently slipping does occur on heavy grades. With the ordinary engine, slipping at such times is a serious matter, as the train is losing speed and may stall on that account after a few repetitions. In the case of the articulated engines, the loss in power by the slipping of one engine is practically gained by the other, in the increase of unbalanced pressure that thereby results. This difference in the unbalanced pressure has the opposite effect on the slipping engine, usually causing it to stop slipping after a few revolutions, without the necessity of closing the throttle. This is explained by the fact that, when the low pressure engine slips, the receiver pressure naturally falls and reduces the back pressure on the high pressure piston, as well as the forward pressure on the low pressure piston; causing the latter engine to stop slipping on account of the friction against the rail under the reduced receiver pressure, which reduction also increases the average unbalanced pressure on the high pressure piston a corresponding amount.

48 When the high pressure engine begins to slip, the receiver pressure increases by the more rapid supply of steam, and with this the back pressure on the high pressure piston is increased, causing this latter engine to resume its grip on the rail and increasing the power of the low pressure engine until the normal power is restored in the high pressure engine.

49 It is further to be noticed, that a simultaneous slipping of both

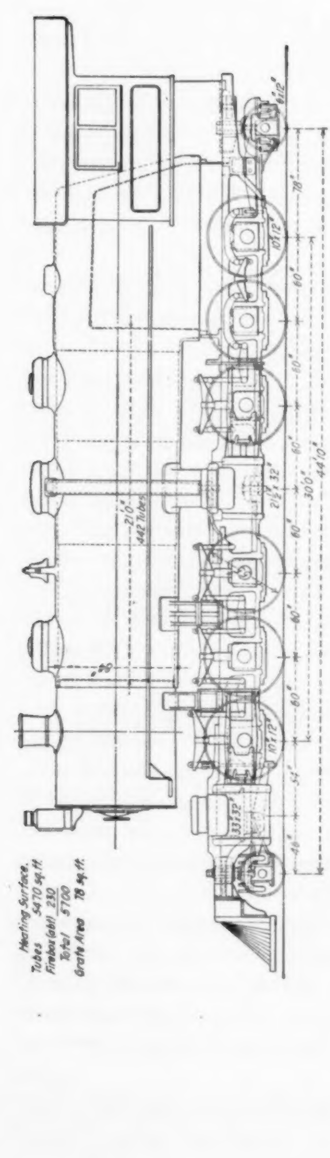

FIG. 17 ELEVATION OF ARTICULATED LOCOMOTIVE FOR THE GREAT NORTHERN
BALDWIN LOCOMOTIVE WORKS, BUILDERS

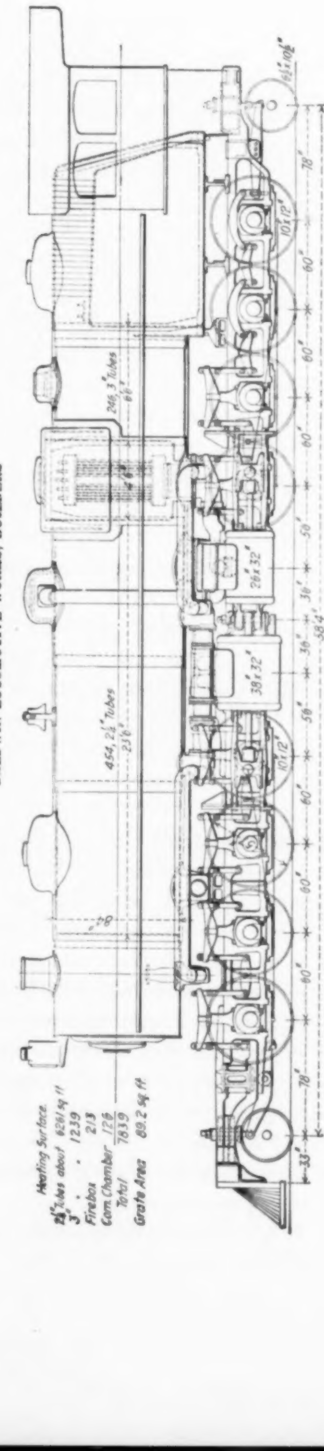

FIG. 18 ELEVATION OF ARTICULATED LOCOMOTIVE BUILDING FOR THE ATCHISON, TOPEKA AND SANTA FÉ RAILROAD
BALDWIN LOCOMOTIVE WORKS, BUILDERS

engines is a very rare occurrence; due to the fact that there is one position of the crank where the turning effort is greater than in other positions, and this is where the slipping generally starts at irregular intervals of revolutions, depending on the condition of the engine as to necessity for repairs. If the wheels are of a very close approximation to the same diameter and running on a straight track, these intervals are longer, because the opposition between the wheels requiring readjustment is less frequent than when the wheels are unevenly worn. In either case, there is the greatest liability for this adjustment in the neighborhood of the greatest turning movement; say every fourth or fifth revolution, on a normal rail condition. Being two independent engines, the coincidence of these conditions is very infrequent. The slipping of one engine may follow that of the other due to the temporary increase in power; but one is seldom found to start slipping, before the other has stopped slipping; and it can, therefore, be said without much exaggeration, that this type of engine is, in effect, a non-slipping engine.

50 When working live steam in all cylinders, generally known as working simple, the slipping is even less perceptible, although over 20 per cent more power is developed; because the live steam supply to the low pressure cylinders and the direct exhaust from the high pressure cylinders are restricted to a very moderate piston speed. From the beginning of the slip, the low pressure piston gets a rapid motion which causes a sudden fall in the pressure and the slip generally stops after a few inches movement of the piston under normal weather and rail conditions. The restricted supply port being fully open, however, the pressure is restored practically simultaneously with the stopping of the slipping.

51 The high pressure engines are not so sensitive; but after a couple of exhausts under slipping, the wheels regain their grip on the rail with comparatively small loss of power and in a period of short duration.

52 The effect on cars and draft gears in starting heavy trains by this type of engine, as well as convertible compound engines on the same principle, is a most important feature, as it is accomplished with a so-called dead pull, without the necessity of taking advantage of the slack in the train with its destructive jerks. These locomotives are, therefore, easier on the draft gears than simple engines of half their size loaded to their full capacity. The reason for this is found in the great starting and emergency power, previously referred to, with which these engines are provided, so that the slack

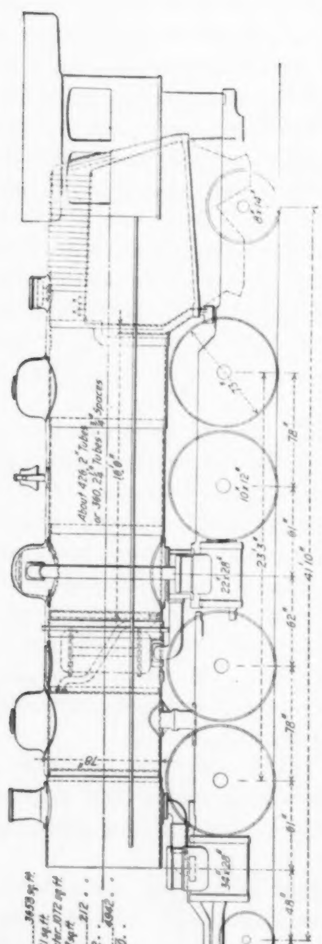

FIG. 19 BALDWIN WORKS' PROPOSED DESIGN OF ARTICULATED PASSENGER LOCOMOTIVE FOR THE ATCHISON, TOPEKA AND SANTA FÉ RAILROAD

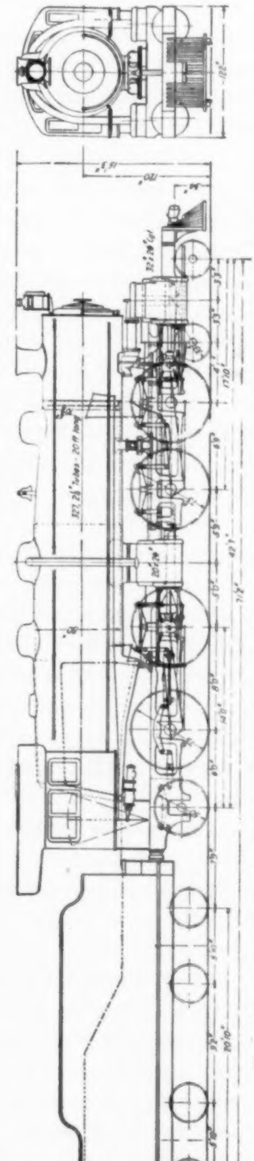

FIG. 20 AMERICAN LOCOMOTIVE COMPANY'S STUDY OF ARTICULATED PASSENGER LOCOMOTIVE

is taken up under very slow speed. This is generally done with light throttles. The front cars start successively under a slight acceleration of the engine, gradually going over to a retardation before the last cars get into motion, after which the engine is given full throttle. In other words the train is stretched first and then it is started under direct pull, so that there need not be any but slight shocks or jerks.

53 These engines are adaptable to a greater variety of conditions than the older types, rendering it possible to double the engine power on a given rail weight; and their advantages are most pronounced as displayed on heavy grades and sharp curvatures.

54 Over a hundred locomotives of this type have been built up to the present date in this country, ranging in weight on drivers from from 106 000 lb. to 410 000 lb. and from 20000 lb. to 125 000 lb. in tractive power. The largest of these is taking the place of three ordinary sized locomotives, in pushing service, and is doing its work very satisfactorily.

55 Their success in freight service will undoubtedly lead to their adoption in passenger service; especially in certain localities to avoid double heading, as the Pacific type locomotive is practically the upper limit for one set of drivers. It, therefore, appears that further development will have to be in the flexible driving wheel base or articulated type.

56 Fig. 19 shows a design of Mallet type locomotive for passenger service, fitted with re-heater between the high and low pressure cylinders and feed water heater in combination therewith, prepared by the Baldwin Locomotive Works for the Atchison, Topeka and Santa Fé Road. Engines of this design are now under construction at the Baldwin Works.

57 A preliminary diagram of a 4442 class articulated passenger locomotive of the American Locomotive Company is shown in Fig. 20. Neither the front truck nor the trailer has any side motion beyond their ordinary axle play because the individual wheel bases would be too short to guide this engine safely, without the rigidity of the truck and trailer. The truck, however, has a rigid center pin around which it can adjust itself in a curve without restriction by the alignment of the engine.

58 It should also be remarked that, due to absence of jerks and slack in starting, as well as the more uniform cylinder pressure, the stresses on the machinery and framework are considerably reduced; and, further, that the milder exhaust produces a less intense heat

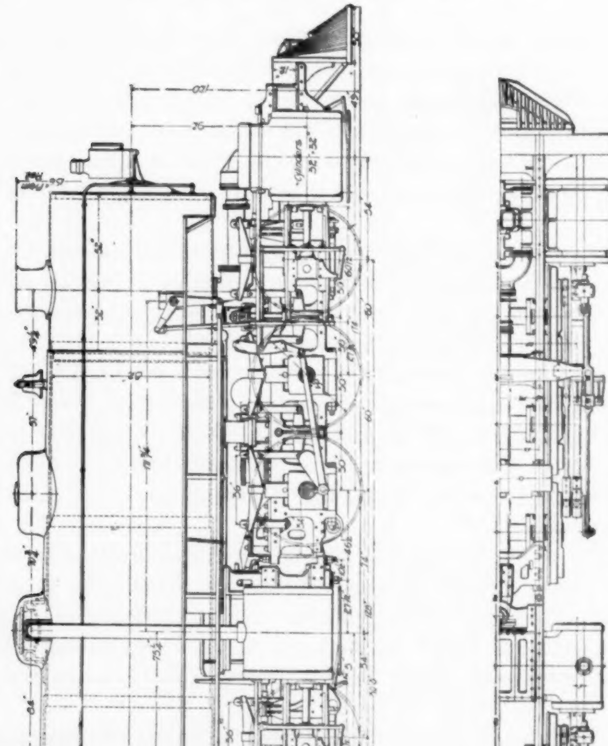

FIG. 21 ELEVATION AND HALF PLAN OF ARTICULATED LOCOMOTIVE FOR BALTIMORE & OHIO RAILROAD

AMERICAN LOCOMOTIVE COMPANY, BUILDERS

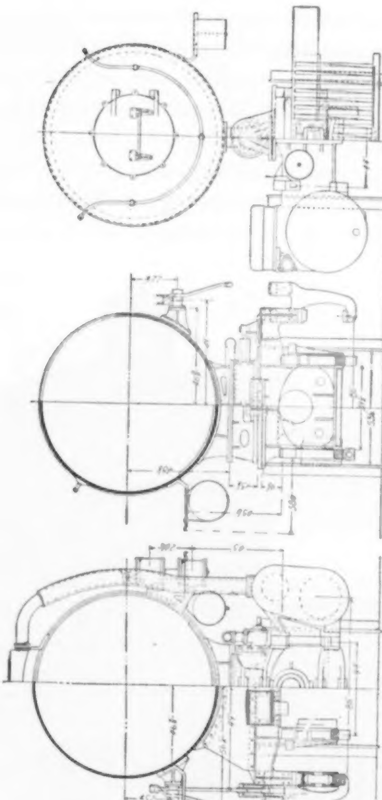

Fig. 22 End Views and Cross Section of Articulated Locomotive for Baltimore & Ohio Railroad
AMERICAN LOCOMOTIVE COMPANY, BUILDERS

and a better utilization of it, all of which contribute to a reduction in the repairs of the locomotive as a whole, compared with a simple engine, if it were practical to construct one of this type. This has never been advanced as a feature to the credit of the articulated engine because it is difficult to give it any definite value; but is referred to as a reply to the often repeated supposition that these engines are hard to keep in repair. As a matter of fact the opposite is the case because on account of the subdivision of the work in two engines the parts are lighter and easier to handle in repairs and renewals.

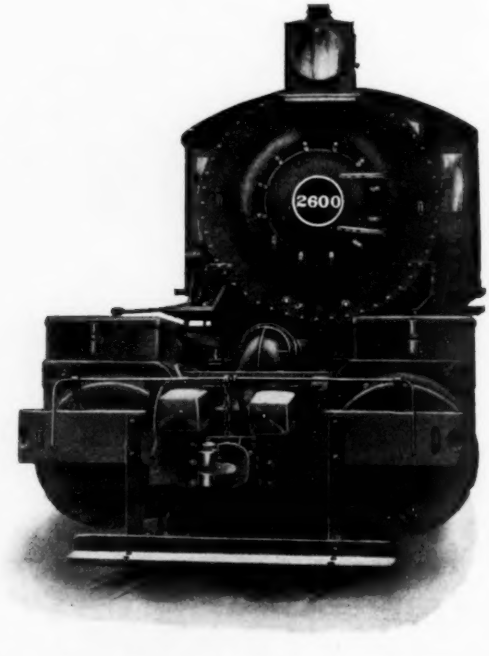

FIG. 23 FRONT END VIEW OF ARTICULATED LOCOMOTIVE ROUNDING A CURVE

59 One of the fundamental principles in locomotive engineering, as applied to conditions as they exist in the United States, is simplicity of construction. This has led to a general reluctance on the part of American railroad officials to accept complications as long as they could avoid it. The tendency toward heavier trains has, however,

made it necessary to supply units of larger hauling capacity than have heretofore been necessary anywhere in the world. For the most exacting freight service, locomotives are required which have entirely outgrown the possibilities of ordinary construction; in very much the same way as marine requirements have outgrown the types of propelling engines, which were entirely satisfactory a generation ago. In fact, the demand for units of large power compels special construction in order that the large units may be operated without damage to the track and structures, and to avoid increasing the size of the moving parts of the locomotive to a prohibitive point.

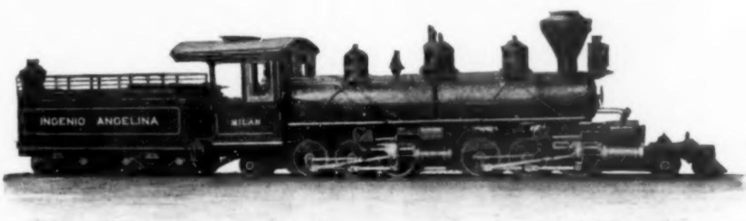

FIG. 24 SMALL ARTICULATED LOCOMOTIVE
BUILT FOR THE INGENIO ANGELINA BY THE BALDWIN LOCOMOTIVE WORKS

Gage	2 ft. 6 in.	Diam. drivers	33 in.
Diam. cyl.	h. p. 10 in. l. p. 15 in.	Wt. on drivers	51 000 lb.
Stroke	16 in.	Total wt.	60 200 lb.
Pressure	180 lb.	Tractive power	11 630 lb.

60 The articulated locomotive represents the highest development in this branch of engineering; and the development is sufficiently important to justify the presentation of the subject in its present stage before a body of engineers who have had much to do with corresponding developments in stationary or marine practice.

61 For the data and illustrations contained in this paper, the writer wishes to express his indebtedness to the American Locomotive Company and to Mr. Samuel Vauclain, General Manager of the Baldwin Locomotive Works.

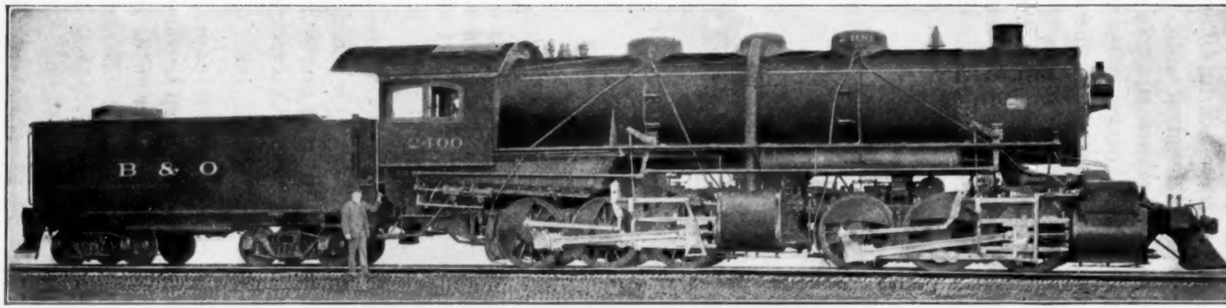

FIG. 25 THE FIRST MALLET ARTICULATED COMPOUND LOCOMOTIVE BUILT IN AMERICA

BUILT FOR THE BALTIMORE AND OHIO RAILROAD BY THE AMERICAN LOCOMOTIVE COMPANY

Gage of track	4 ft. 8½ in.	Outside diameter of tubes	2¼ in.
Diameter of cylinders	h.p. 20 in.; l.p. 32 in.	Length of tubes	21 ft.
Stroke of piston	32 in.	Driving wheel base	30 ft. 8 in.
Diameter of driving wheels	56 in.	Rigid wheel base	10 ft.
Outside diameter of boiler at front end	84 in.	Total weight in working order	334 500 lb.
Working pressure	235 lb.	Weight on driving wheels	334 500 lb.
Length of firebox	108 in.	Tractive power	71 500 lb.
Width of firebox	96 in.	Tractive power working simple	91 000 lb.
Number of tubes	436	Factor of adhesion	4.67

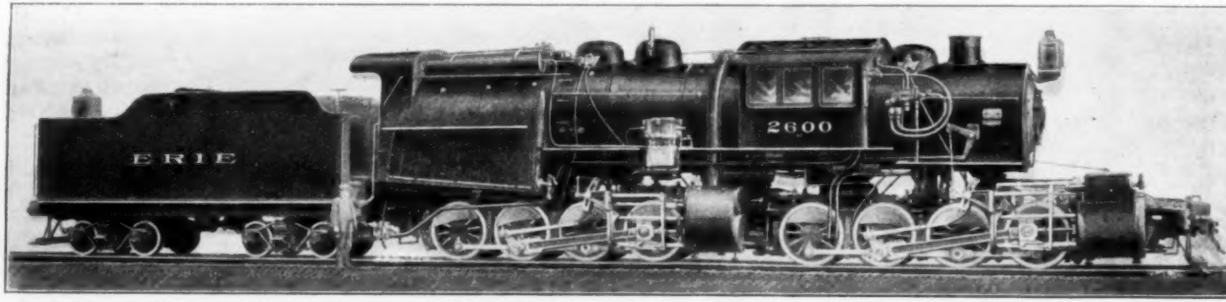

FIG. 26 HEAVIEST AND MOST POWERFUL LOCOMOTIVE IN THE WORLD

MALLET ARTICULATED COMPOUND LOCOMOTIVE BUILT FOR THE ERIE RAILROAD BY THE AMERICAN LOCOMOTIVE COMPANY

Gage of track	4 ft. 8½ in.	Outside diameter of tubes	2¼ in.
Diameter of cylinders	h.p. 25 in.; l.p. 39 in.	Length of tubes	21 ft.
Stroke of piston	28 in.	Driving wheel base	39 ft. 2 in.
Diameter of driving wheels	51 in.	Rigid wheel base	14 ft. 3 in.
Outside diameter of boiler at front end	84 in.	Total weight in working order	410 000 lb.
Working pressure	215 lb.	Weight on driving wheels	410 000 lb.
Length of firebox	126 in.	Tractive power	94 800 lb.
Width of firebox	114 in.	Tractive power, working simple	120 000 lb.
Number of tubes	404	Factor of adhesion	4.33

This engine is equipped with boiler fitted with combustion chamber.

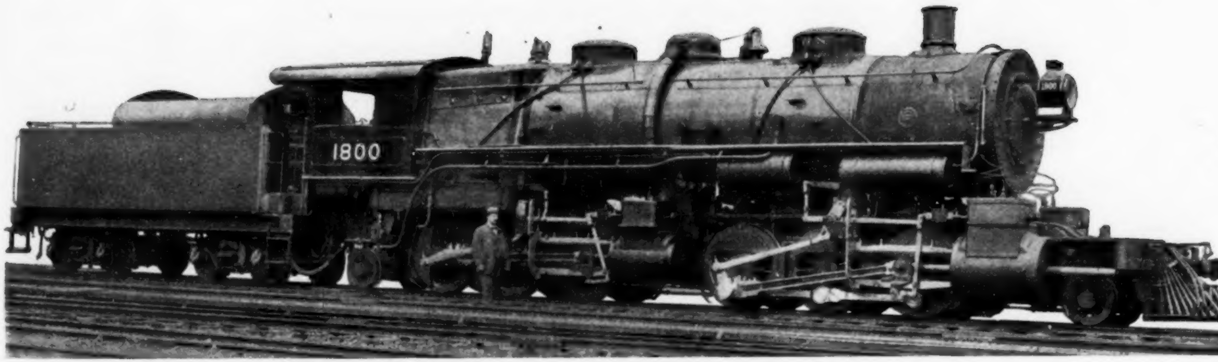

FIG. 27 MALLET ARTICULATED COMPOUND LOCOMOTIVE BUILT FOR THE GREAT NORTHERN RAILWAY

BY THE BALDWIN LOCOMOTIVE WORKS

Gage of track	4 ft. $8\frac{1}{2}$ in.
Diameter of cylinders	h.p. $21\frac{1}{2}$ in.; l.p. 33 in.
Stroke of piston	32 in.
Diameter of driving wheels	55 in.
Outside diameter of boiler at front end	84 in.
Working pressure	200 lb.
Length of firebox	117 in.
Width of firebox	96 in.
Number of tubes	441
Outside diameter of tubes	$2\frac{1}{4}$ in.
Length of tubes	21 ft.
Driving wheel base	30 ft.
Rigid wheel base	10 ft.
Total weight in working order	355 000 lb.
Weight on driving wheels	316 000 lb.
Tractive power	71 600 lb.
Factor of adhesion	4.41

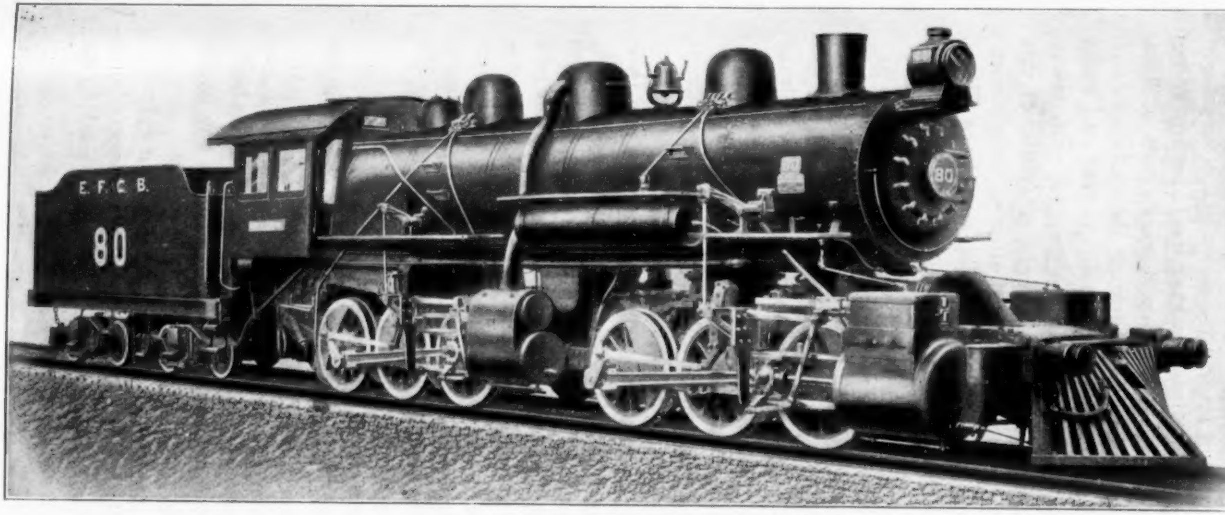

FIG. 28 MALLET ARTICULATED COMPOUND LOCOMOTIVE BUILT FOR THE CENTRAL RAILWAY OF BRAZIL

BY THE AMERICAN LOCOMOTIVE COMPANY

Gage of track	5 ft. 3 in.	Outside diameter of tubes	2 in.
Diameter of cylinders	h.p. 17½ in.; l.p. 28 in.	Length of tubes	18 ft.
Stroke of piston	26 in.	Driving wheel base	27 ft. 8 in.
Diameter of driving wheels	50 in.	Rigid wheel base	9 ft.
Outside diameter of boiler at front end	64 in.	Total weight in working order	206 000 lb.
Working pressure	200 lb.	Weight on driving wheels	206 000 lb.
Length of firebox	90 in.	Tractive power	42 400 lb.
Width of firebox	65 in.	Tractive power, working simple	52 000 lb.
Number of tubes	234	Factor of adhesion	4.86

DISCUSSION

Mr. F. J. Cole The type of locomotive described is singularly well adapted to a wider range of service than perhaps any other design. It was originally intended for narrow-gage roads of light construction built for military purposes, following the undulations of the country without grading, necessitating sharp curves and steep grades, in combination with light rails and the greatest economy in the construction of track, bridges, etc. The characteristics of this design, namely, flexibility and uniform distribution of weight combined with the use of two separate engines which would not slip at the same time, rendered the design very attractive for these narrow-gage railroads.

2 The first engine of this class was built about the year 1887 and at the present time there are approximately 500 running in Europe. Some five years ago the Mallet locomotive was introduced in this country and its merits and efficiency have been recognized by railroad men wherever they have been put in service. The three articulated engines designed and constructed for the Erie Railroad for use on the Susquehanna grade, of the 0880 type, are the heaviest and most powerful locomotives ever built and operated.

3 When we compare the early locomotives of this type and their work on light narrow-gage railroads, with those built last year for the Erie Railroad, where each engine took the place of three standard 100-ton consolidations, we at once appreciate the great range to which this general class of articulated locomotives is adapted. A locomotive must possess peculiar inherent qualities to enable it to perform so satisfactorily such a wide range of service, from two-foot narrow-gage roads to main line work on leading trunk lines, utilizing the extreme limits of axle loads and adhesion.

4 Some of these distinctive features are:

- *a* Short rigid wheel base, ranging from about 2 ft. 8 in. to 14 ft. 3 in.
- *b* Rigid connection of high-pressure steam-pipes to the rear engine, permitting steam connections between the boiler and cylinders in the ordinary way, without the use of flexible connections.
- *c* Flexible steam connections to the low-pressure cylinders, which are easily kept tight, as a receiver pressure of only about 70 lb. has to be provided for and the packing for this pressure is not a difficult problem.

d Extreme flexibility; forward group of driving wheels, operated by low-pressure cylinders, swing freely when passing curves.

e The articulated locomotive is essentially a compound proposition presenting many advantages over the use of four simple cylinders. These engines cannot be designed as simple engines and work as satisfactorily as the present arrangement of compound cylinders, because this would necessitate the use of high-pressure flexible steam pipes and introduce other complications.

f The total weight being carried on the drivers, makes it possible to use a very high tractive power, especially valuable for pushing and helping service.

g Impossibility of both engines slipping at the same time, the receiver pressure automatically taking care of this. If the high-pressure engines commence to slip, the receiver pressure will be increased and produce greater back pressure on the high-pressure pistons, and consequently greater pressure will be admitted to the low-pressure pistons, resulting in alternating slipping from one engine to the other. This is of great value in hill climbing.

5 Considering the matter of axle-loads, this type presents many advantages because from four to eight driving axles can be employed. The limitations of the consolidation or 280 class are:

200 000 lb.	on	drivers,	50 000 lb.	axle	load	
220 000	"	"	"	55 000 "	"	"
240 000	"	"	"	60 000 "	"	"

In the decapod or 2-10-0 type we have these limitations:

250 000 lb.	on	drivers,	50 000 lb.	axle	load	
275 000	"	"	"	55 000 "	"	"
300 000	"	"	"	60 000 "	"	"

6 The use of 2-10-0 type, however, is very questionable except with small wheels on account of the extremely long rigid wheel base, with its necessary accompaniment of extreme flange-wear and friction in passing any but the longest radii curves. With the Mallet articulated engine a much wider range of driving wheel loads is permissible. The 0660 type permits the following loads:

300 000 lb.	on	drivers,	50 000 lb.	axle	load	
330 000	"	"	"	55 000 "	"	"
360 000	"	"	"	60 000 "	"	"

The 0880 type permits the following weights on drivers:

400,000 lb. on drivers,			50,000 lb. axle load
440,000 "	"	"	55,000 " " "
480,000 "	"	"	60,000 " " "

7 The above figures show the possibilities of building locomotives of enormous tractive power larger than that possessed by any other type. The wide range of permissible axle-loads and the great flexibility, in combination with the economy produced by compounding, especially in slow service, all tend to make the design particularly well adapted to a wide range of work.

8 The ease with which these engines are fired is a matter of general surprise. This is largely due to the use of compounding, which reaches its maximum efficiency at slow speeds and long cut-offs, whereby the work of the fireman is materially reduced.

9 In ordinary service, especially for helping and pushing, the entire weight is needed for adhesion and no useful purpose is served by the additional complication of leading truck wheels. No sharp flanges have developed on the Baltimore & Ohio 0660 type after four years of service. This locomotive has been operated 24 hours per day pushing up hill and backing down over sharp curves without detrimental effects; while the ordinary consolidations in use on this road do wear their flanges badly. This proves conclusively the extreme flexibility of this type and its ability to move freely around curves without the use of guiding wheels. The ease with which a six-wheel Pullman car truck passes around curves, presenting an almost ideal construction for this purpose, explains this freedom from flange wear. It can readily be seen that the front low-pressure engine of the Mallet locomotive of the 0660 type is really nothing more than a large six-wheel truck and it seems quite as unnecessary to add leading truck wheels to an engine of this kind for ordinary conditions as to add a leading wheel to the six-wheel car truck.

10 The sliding support carrying the weight of the boiler is located in a position to equalize wheel loads, and by swinging links between the front of the rear frames and the back of the front frames, extreme refinement can be made in the equalization of weight, so that the wheels may be equally loaded. The employment of leading or trailing truck wheels does not seem to be necessary except for the possible use in road service where speeds exceed 35 and 40 miles an hour and the boiler requirements are such as to make it impossible to utilize the entire weight of the engine for adhesive purposes. The only justification for their use seems to be in cases where the extreme boiler capacity

is required. In the operation of trains at good speeds, the boiler capacity is the chief requirement and the tractive power a secondary consideration; as in the case of Atlantic or Pacific engines, where a great deal of adhesive weight is sacrificed for the purpose of obtaining greater boiler capacity.

11 Two years ago, in discussing Mr. J. E. Muhlfeld's paper on large electric and steam locomotives before the New York Railroad Club, I called attention to some advantages of this design for heavy service, among which were the following:

a Ease of turning the engine around on a "Y" of 23 deg. with no grinding of flanges and almost entire absence of flange friction, the engine appearing to pass the curves as easily as a car with the ordinary forms of trucks.

b Absence of any unusual exertion on the part of the fireman in maintaining steam pressure.

c Ease of operation, the power reverse gear making much less exertion, than in the ordinary two-cylinder engine, necesary to reverse or alter the cut-off, which involves moving four valves.

12 The question of guiding trucks was also considered, and the opinion which I found at that time was that their use was unnecessary and their only effect would be to complicate the design without serving any useful purpose. The two years which have elapsed since that time have merely tended to confirm this opinion and the absence of flange wear on the front driver at the present time bears testimony to the inherent flexibility of this design for use on roads having sharp curves.

Mr. Harrington Emerson A few years ago Mr. B. J. Arnold, a prominent electrical engineer, stated that there was no known way of moving freight so cheaply as by putting a steam locomotive ahead of a train. During the last few years there has been an attempt by electrical engineers to rush the subject, and to pretend that the steam locomotive is obsolete. They proved their case by assuming ideal conditions for the installation of electric traction, assuming at the same time that current locomotive practice is the best obtainable.

2 Two things have occurred in recent times to put far into the future this electrification of all railroads. One is the panic, by which railroad managers have been reminded that it would be expensive

to depreciate and make obsolete all their existing power as well as round-houses and division points: the other factor is the Mallet type of engine.

3 Before the Mallet was developed, locomotives had been lengthened out until we had for mountain grades the famous Santa Fé type, with its 34-ft. wheel bases. That locomotive operated very economically, carrying loads for less than any electric installation could possibly have carried them.

4 I remember a discussion between a former superintendent of motive power, and the general manager of the Santa Fé system, in which the latter accused the superintendent's locomotive of spreading the track so that nothing could run over it. The superintendent replied that it was the business of the general manager to keep the track in order. The question of the wheel base, which was the limiting question in the old type of locomotive, is obviated in the Mallet type, so that for many years the Mallet will be found operating with the highest economy on mountain grades.

5 There is a difficulty with the Mallet that was also experienced with the Santa Fé type. I was once with the superintendent when he received a telegram that one of the locomotives was off the track. He wired to find what speed she was running at, and the reply came that the locomotive was not running at all, but was standing still. The superintendent remarked, "We are up against it when locomotives jump the track when standing still." That difficulty may also be experienced from the heavy weight of the Mallet on drivers.

Mr. L. R. Pomeroy The paper and discussion thus far have been confined mainly to questions of design and construction. The speaker desires to call attention briefly, to some of the commercial advantages of the Mallet type.

2 Generally speaking, the gain to be expected, from the substitution of electric for steam operation, depends greatly on the density of traffic coupled with the frequency of units. And further, if it is not possible to accomplish by electric service something now impossible with steam service, then the adoption of electric service is not commercially practical, for there is nothing to be gained, *per se*, by the mere substitution of one kind of power for another.

3 This at once suggests the question of capacity. On certain mountain grades where the maintenance of expensive helper service is necessary, about all that can be figured out in favor of electric haulage, owing to the limited volume of business in the particular case, would

be that the tonnage per train on a given grade or section could be nearly doubled, or the train mileage halved, without a corresponding increase in train-crew expense.

4 For instance, a 50-mile mountain section having a maximum grade of 2.2 per cent, with seven trains per day in each direction. The reduction of one-half in train mileage with the same tonnage at 50 cents per train mile, the rate covering the items directly affected as used in computing the saving or advantages in grade reduction, would amount to \$65,000 per annum, and this amount capitalized at 6 per cent would equal about \$1,000,000. But to obtain this saving electrically the complete electric apparatus would probably cost considerably more than this capitalized amount, whereas the requisite number of Mallet compound steam locomotives to perform the service would cost about one-third of the amount necessary for an equivalent electric service.

5 To state the case another way, i.e., based on train-crew saving only, leaving out entirely all other advantages. The total of 14 trains is the equivalent of about 700 train miles per day and the cost for train crews amounts to from 12½ to 15 cents per train mile. Then the saving per annum will be

$$\frac{700 \times 0.14 \times 365}{2} = \$17,800$$

This amount capitalized at 6 per cent equals about \$300,000, or more than enough to pay for the required number of Mallet locomotives to perform the service.

6 The foregoing is not meant to be a reflection on electric possibilities, *per se*. The particular case cited does not possess the inherent magnitude of business to warrant or justify an electric proposition, but it does show that until future conditions change there are a great many cases where the Mallet type of locomotive will serve as a profitable bridge over the deep chasm between present conditions and the eventual supremacy of the electric locomotive.

MR. GEORGE L. FOWLER As I was coming up the stairs a few moments ago I met Professor Goss, who was about to leave the building, and he asked me to say a word for him. He wanted me to call attention to the fact that when Mallet designed his locomotive, it was designed for narrow gage roads, and for comparatively light traffic, as has already been mentioned in the discussion, and when an attempt was made to apply this principle to the tremen-

dously heavy locomotives required for American service, such as we have in the Great Northern, Baltimore & Ohio and Erie engines, the problem assumed a magnitude which Mallet probably did not consider when he first designed his engine, so that in taking that principle and adapting it to the heavy locomotives of American practice, great merit must be given to the designer of the first heavy Mallet engine in this country, the gentleman who presented the paper to you this afternoon, Mr. Mellin. This is for Professor Goss.

Mr. George R. Henderson The only thing I want to note is the necessity in these large engines for a pretty liberal supply of fuel, and possibly a liberal supply of men to put it in, unless we use liquid fuel, or have some mechanical means of putting coal in the fire-boxes. Take one of the Mallet engines with 80,000 lb. tractive force, and attempt to use that force at a speed of anywhere near 15 miles an hour; a large amount of coal, five tons, or 10,000 lb., an hour, would be called for in the fire-box, and I think it is time for those interested in mechanical stokers to suggest a practical means of supplying this fuel to the engines, so that they may not merely pull cars at slow speed, but give a fairly good return in horse power developed for money invested.

Mr. Alfred Lovell The Mallet articulated locomotive presents an opportunity to utilize steam by double expansion without increasing the number of working-parts over those required for single-expansion engines of corresponding weight and power, and to increase further the economy thus secured, by the introduction of superheating; an advantage touched upon only lightly by the author, and deserving of a more complete recognition.

2 The key-note of present-day engineering thought and activity is the conservation of resources, as was impressed upon us by the valuable address of the President of the Society at the opening of the December meeting.

3 In this address it is stated that under the best conditions that prevail in stationary power plants not more than 15 per cent of the heat value of the coal used is utilized in actual work produced at the engine, and that under average conditions a considerably smaller percentage is realized. The best conditions referred to evidently contemplate the multiple expansion principle in the use of steam. In the best locomotive practice the conditions are still less advantageous for the utilization of the heat value of fuel, and in average

locomotive service, the percentage is probably much less than in average stationary plants. In the locomotive tests at the St. Louis Exposition in 1904, the work at the draw bar, not including engine friction, represented a percentage of the heat value of the coal ranging from about $3\frac{4}{10}$ per cent in simple engines at high speed, to about $8\frac{6}{10}$ per cent in the best compound engines at low speed.

4 Good engineering practice thus demands that locomotives be so designed as to increase to a maximum the utilization of the heat of the fuel, and without excessive complication that might result in excessive maintenance cost, or delays to service.

5 In Europe the advantage of double expansion in locomotive practice has long been recognized. In American railway service, the last twenty years have demonstrated absolutely that the compound use of steam in locomotive service results in a substantial saving of fuel, the percentage ranging from 5 to 40 per cent, according to the character and conditions of service. It has also been demonstrated that in long continued service, covering several years, boiler repairs are less with the compound, due to less demand for steam and milder exhaust.

6 When the compound locomotive was first built with four cylinders and their complement of pistons, piston rods, and rod and valve packing, it was found that, notwithstanding, the decreased boiler wear, the liability to disorder from breakage or wear of engine parts, on account of the slightly increased number of wearing parts, sometimes quite offset the advantage of fuel economy. The cross compound or two-cylinder type, high-pressure on one side and low-pressure on the other, was expected to overcome this objection, and these were built in considerable numbers for several years succeeding 1895.

7 Experience proved that under certain conditions this latter type developed unequal or twisting strains in the opposite sides, sometimes resulting in breakage of frames and other parts. Moreover, the demand for increased size of units of power soon brought the low pressure cylinder in the two-cylinder compound to the limit of size permissible. The result was a further development of four-cylinder compounds, and the tandem and balanced compound types were produced.

8 Both of these types are good fuel savers, and are free from some of the objectionable features of the earlier types, while the tandem gives opportunity to construct units of great size and power. Yet the slightly greater number of working parts, and the anticipated

or actual difficulty of reaching some of these parts in making running repairs, cause many to reject them and to perpetuate the wasteful method of using the steam single expansively.

9 The articulated compound locomotive bids fair to overcome these objections entirely. This locomotive is in effect two engines with one large boiler and one fire-box. The two engines together use the steam double expansively, and with no more cylinders and working parts than two single-expansion engines. Boiler repairs should be relatively small since the minimum of steam is required and the exhaust is mild. Wearing parts of engines are easily accessible.

10 One engineer operates both engines as readily as one four-cylinder engine. Owing to the economical use of steam, one fireman can stoke a locomotive of this type having very large tractive power. The type thus provides means for the economical use of fuel with a minimum of labor and no increase of disorder delays, or in engine repair cost, and with a probable reduction in boiler repair cost as compared with single-expansion locomotives of corresponding power.

11 Superheating of steam in locomotive practice has been shown to effect fuel economies in single-expansion locomotives nearly or quite equal to those secured by compounding. It is probable that with proper arrangements a combination of double-expansion, and of superheating the steam before using, or reheating it between the high and low pressure cylinders, will give greater fuel economy than either compounding or superheating alone. The flexibility of arrangement possible in this type of locomotive presents a most advantageous opportunity for this combination.

12 The necessities of the future will demand in locomotive practice the greatest possible conservation of skilled labor and of fuel, and for the reasons mentioned the Mallet articulated compound locomotive is worthy of the spirit of the day, and will perform an important part in the economies of the future.

Mr. S. M. Vauclain exhibited a large number of lantern slides of which he gave an interesting running account by way of discussion of the paper. These related to the initial work of Mallet in the development of the type of locomotive that bears his name and to locomotives of this type built by the Baldwin Locomotive Works. He said: It is not my purpose to criticise the admirable paper of Mr. Mellin; but merely to preface my discussion with remarks to

the effect that I do not coincide with the author in all his conclusions or in his treatment of the principles upon which Mr. Mallet has spent so many years of his most useful career. One design of this most flexible type of locomotive may be so treated as to include many devices considered entirely unnecessary by the inventor, yet on the other hand it is quite natural to make additions or changes common to American practice and suggested by years of experience on American roads. It is not my purpose to discuss the details of the paper but to add what is most lacking, a historical sketch giving the fullest honor to the original inventor, and to show the development of the subject in the United States from the standpoint of our progressive railroad men.

2 It was Mallet who, in 1875, started the era of economics in locomotive building destined to keep all minor lights in locomotive building fully occupied. He did this by means of a two-cylinder or cross-compound locomotive of Roentgen type built at Creusot for the Bayonne & Biarritz Railway. In 1877 he recognized the inefficiency of this type, since proved in America, and adopted the more sensible four-cylinder tandem type. But so rapidly did his mind work that during the same year he changed from the tandem type to one in which the cylinders were coupled to separate systems of wheels and operated independently, but with one supply of steam. His idea was either to use a rigid frame or to articulate. The De-Glehm engines, the first of which was built in 1885 for the *Chemin de Fer du Nord* and later for the Paris, Lyons & Mediterranean, were of the former or solid-frame pattern. He shortly thereafter successfully introduced the articulated frame which is considered a characteristic feature of the type which now bears his name, and we owe to Mr. Mallet all the honor that may ensue from its introduction in this country. It is to be regretted that, like Walschaerts, Mallet is unable to realize the full benefit of his inventions, owing to the time-limit of the patent law.

3 The Baldwin Locomotive Works recognized the merit of Mallet's invention and in 1889 made a careful investigation of the engine used by the Decauville Railway at the Paris Exposition. Numerous efforts were made to interest both foreign and domestic railroads, and Mr. Mallet personally assisted in planning the general types. The first design for a domestic road was submitted to the Erie Railroad in 1898, and it was at that time considered a very heavy locomotive.

4 Many designs were worked out for J. W. Kendrick, Second

Vice-President of the Atchison, Topeka & Santa Fé Railway. The later designs of this series had an intermediate chamber to overcome the objectionable length of tubes, fire-box combustion chamber, and leading and trailing trucks. In one of the designs for passenger service it was intended to produce a locomotive, with superheater, of sufficient power for the heaviest train on grades of two per cent and less.

5 The design finally proposed by Mr. Kendrick had a reheater between the high and low pressure cylinders through which the steam must pass on the way to the low-pressure engine, and also a feed-water heater. The cylinders in this engine face each other as in previous design for freight service.

6 Designs were proposed for freight service, embodying the jointed boiler feature having cylinders facing each other, open combustion chamber covered by a movable cap, etc. The design finally adopted for freight service for this railroad had a reheater, superheater and feed-water heater, and a detachable front section making all parts readily accessible. The final type agreed upon for passenger service has five pairs of driving wheels, a four-wheel leading truck and a two-wheel trailing truck.

7 One of the first to recognize the merit of the Mallet type of locomotive was J. J. Hill, who ordered five of these locomotives. A design of a locomotive without a truck was first submitted, but he demanded a trailing, as well as a leading, truck, and experience with this arrangement has proven his diversion from previous practice to be justified. This was also in accordance with our own preference. On the Great Northern Railway, 67 engines of the Mallet type are now in use working on grades of from 0.6 per cent to 2.2 per cent.

8 The speaker had received from Geo. H. Emerson, Superintendent of Motive Power of this road, data upon the performance of some of these engines, which he submitted in abstract. These are given in full under Mr. Emerson's name in the discussion immediately following.

9 Engines designed for service on the Mexican Central were then referred to, and a résumé was given of designs proposed for the Southern Pacific and worked out at the solicitation of J. Kruttschnitt. The first design had no trucks, no tender side tanks being employed. The side tanks, however, reduced the boiler capacity and the coal box was inadequate. The second design provided for separate cabs for the engineer and the fireman. The engineer was placed forward and the fireman in the rear. A combustion chamber

was employed in the third design in order to reduce the length of flues to a reasonable figure. The fourth design was made to overcome the objection of small driving wheels; in it were employed all the possible details of the Harriman common standards. This design shows a logical outcome, and two locomotives of the type are now in the course of construction. The low-pressure cylinders are detachable, the boiler is separable and the steam from the high to the low pressure cylinders passes through a Vauclain reheater.

10 In closing, the speaker referred to a special design of his own, differing from that of the Mallet type previously noted, in having a flexible boiler instead of provision for sliding contact for the front end of the boiler to the supporting member of the frame, providing the necessary lateral movement in rounding curves. The boiler is in two sections, each firmly secured to its own cylinders, and frames hinged after the Mallet design, but the two sections of the boiler joined by a flexible connection between the two smoke chambers containing the superheater and the reheater.

MR. G. H. EMERSON In October 1906 the Great Northern Railway received five large Mallet engines, the two high-pressure cylinders being $21\frac{1}{2}$ in. in diameter by 32 in. stroke, the two low-pressure cylinders located on forward engine 33 in. in diameter by 32 in. stroke: boiler pressure 200 lb.; weight on drivers 316,000 lb.; total weight of engine and tender loaded 503,200 lb.; diameter of drivers 55 in.; engine having two-wheel pony truck, three pairs of drivers on front engine with three pairs of drivers followed by two-wheel trailer on rear engine; boiler having total heating surface of 5700 sq. ft. and grate area of 78 sq. ft.; maximum rigid wheel base on either engine, 10 ft. This engine will be referred to hereafter as class L-1.

2 The largest engines heretofore used by this company were the consolidation F-class having cylinders 20 by 32; carrying 210 lb. boiler pressure; 180,000 lb. on driver; total weight of engine and tender 318,000 lb.; total heating surface of boiler 2768.4; total grate area 59.2; diameter of drivers 55 in.

3 The five large class L-1 Mallets above referred to were purchased as helpers on heavy grades and were placed in service on the Cascade Mountains between Skykomish and Leavenworth, where the ruling grade is 2.2 per cent. East from Skykomish there are 22 miles of continuous 2.2 per cent grade with a let-up through the Cascade Tunnel of 1.7 for a distance of three miles. West from Leavenworth there is almost a continuous grade of 2.2 per cent for

a distance of 32 miles ending at Cascade Tunnel station. The engines were put to work on this hill in helper service to take the place of the consolidation F-8 type as pusher and to help an F-8 engine used in road service between Leavenworth and Delta, a distance of 109 miles.

4 The engines were so large they could not be taken care of with the facilities provided. As they could not be turned around without providing further facilities, they were designed with pony truck in front and trailer behind and during the first winter's service, therefore, they were not turned nor were they taken into a roundhouse.

5 Soon after receiving these engines tests were made to determine the tonnage which they could handle, and their economy as compared with the consolidation engine. One L-1 was first used as a helper engine with an F-8 as a road engine and it was found that the two engines could easily handle 1300 tons up the mountain. It was further developed that while the F-8 handled only 500 the L-1 handled 800 tons of the train, using no more coal with this load, and in fact somewhat less, than the consolidation with her part of the train.

6 The first winter's performance was so favorable to the Mallet engine, both from an operating and a maintenance standpoint, that arrangements were made for additional Mallets, as it had been decided they could be used in road service as well as on hill service, and in May 1907 25 road Mallets were received, somewhat smaller than the first ones purchased, their high-pressure cylinders being 20 by 30; low pressure 31 by 30, 200 lb. boiler pressure; 250,000 lb. on drivers; total weight of tender and engine 451,000; total heating surface 3914; grate area 53.4 sq. ft. The general design of this engine was the same as that of the former one, except that it was of smaller dimensions, and the boiler provided on the engine was identical with the boiler used on Prairie and Pacific engines having cylinders 22 by 30 with 69-in. drivers, carrying 210 lb. steam pressure. This last or road class will hereafter be designated as L-2.

7 These engines were put in service on a section of the road where there is a 1 per cent grade and showed themselves so economical that in May 1908 further Mallet engines were received as follows: 17 type L-1 and 20 type L-2, it having been decided to extend the use of the Mallet engines to districts having grades as low as 0.72 per cent.

PERFORMANCE ON CASCADE DIVISION

8 Distance from Leavenworth to Everett, 108.70 miles; ruling

grade, both directions, 2.2 per cent; total ascent: west, 2192, east, 3376.

9 On the main line of the Cascade Division between Leavenworth and Everett, the Mallet class L-1 engine has superseded the consolidation F-8 class. As explained in Par. 3, the large Mallets were first introduced on the hill between Skykomish and Leavenworth with consolidation train engine and L-1 helpers used on the hill only. Up to the present time the tonnage taken over this mountain has been gradually increased from 1050, with two consolidation engines, to 1600 tons now being hauled with L-1 engines. The L-1 engines have replaced the consolidation engines and it is now the practice to start out from Everett with one L-1 engine used as a road engine taking 1600 tons as far as Skykomish over a ruling grade of one per cent. At Skykomish another L-1 is put on as a pusher and the two engines take the 1600-ton train over the mountain. The tonnage hauled in the opposite direction is the same and the L-1 or large Mallet helper has proved herself not only worthy of helper service, but a good reliable road engine, and the combination of road and helper service works out admirably on this division, making it unnecessary going east to reduce the tonnage at Skykomish in order to get over the heavy grade.

10 Recent performance shows that on a round trip over this division the L-1 engines handled 1600 tons with a total of $43\frac{5}{6}$ tons of coal or equivalent to 25.13 lb. of coal per 100-ton mile. The F-8 consolidation type could handle only a 1050-ton train, with practically the same amount of coal, or equivalent to 38.29 lb. of coal per 100-ton mile. In other words, the tonnage on this division has been increased at least 52 per cent with a saving of 34.39 per cent lb. of coal per 100-ton mile, due to the Mallet engine. The pusher engines on this district are allowed only switching mileage, that is, 6 miles per hour while in actual service, and for this reason it is somewhat hard to make a comparison with other engines, as they will take the 1600-ton train over the mountain on the hardest pull at a speed of not less than 8 miles per hour.

11 A good showing in repairs is made by the average of 22.33 cents per mile for the year ending June 30, 1908, especially when it is considered that during this period the engines were exclusively in pusher service. Since putting L-1 large Mallets in road service to replace all the consolidation type, the performance has been so satisfactory that only four are now used as pushers exclusively, two as helpers over the Cascade mountains and two on the Butte Division in transfer service.

PERFORMANCE ON SPOKANE DIVISION

12 Distance from Spokane to Leavenworth: 197.40 miles; ruling grade, both directions, 1.0 per cent; total ascent: west, 1351, east 2186.

13 The 1600 tons delivered from Cascade Division at Leavenworth are here reduced to 1450. A small Mallet L-2 takes this train to Hillyard, a distance of about 202 miles. Here too the hauling capacity of the Mallet engine has made possible an increase in tonnage from 1100 tons hauled by the consolidation to 1450 tons, an increase of 31.8 per cent. The run is so long that this tonnage of 1450 has been established by this division in order to get trains over the district in a reasonable time. Engines have not been loaded down to mere drags, but the L-2 Mallet engine handles this tonnage, from 8 to 10 miles per hour on the heaviest hills, up to 30 miles per hour over portions of the district where the grade is not so hard. The performance for the year ending June 30, 1908, shows 22.04 lb. of coal per 100-ton mile on this district, a saving of 27.5 per cent over a consolidation.

PERFORMANCE ON KALISPELL DIVISION

14 Distance from Cutbank to Whitefish, 127.88 miles: ruling grade: west, 1.0, east, 1.8 per cent; total ascent: west 1613, east 2305.

15 On the Kalispell Division, which is the next division east where Mallet engines are used, the L-2 engine takes a train of 1700 tons from Whitefish to Essex, where the ruling grade is 0.8 per cent. At Essex a large L-1 helper is put on to help the train up to Summit, a distance of 18 miles, where the ruling grade is 1.8 per cent. West bound, L-2 engine takes a train of 1450 tons from Cutbank to Whitefish, ruling grade being 1 per cent. On this district replacement of the F-9 class consolidation by the Mallet engine has increased the tonnage 20 per cent with a corresponding reduction in coal per 100-ton mile of 20.7 per cent.

PERFORMANCE ON MONTANA DIVISION

16 Distance from Havre to Cutbank, 129.37 miles; ruling grade: west 1.0, east 0.8; total ascent: west 1952, east 712.

17 From Cutbank to Havre, another L-2 Mallet engine carries through the 1700 tons delivered from the Kalispell Division. Going

west from Havre, the L-2 road Mallet handles 1450 tons. The round trip on this division with L-2 engine is made with 32 tons of coal, or equivalent to 15.75 lb. of coal per 100-ton mile. F-7 consolidation previously used, requiring the same amount of coal, handled only 1200 tons west and 1425 east, or equivalent to 18.9 lb. of coal per 100-ton mile, showing a decrease by the use of the Mallet engine of 16.6 per cent of coal per 100-ton mile, and an increase of 20 per cent tonnage.

PERFORMANCE ON MINOT DIVISION

18 Distance from Minot to Williston, 120.83 miles; ruling grade both directions, 0.72 per cent; total ascent: west 1069.7, east 777.1.

19 Here the L-2 small Mallet has replaced the F-8 consolidation, having increased tonnage from 1600 to 2200. The L-2 engine makes the round trip on 30 tons of coal, or equivalent to 11.29 lb. of coal per 100-ton mile as against 15.49 lb. of coal per 100-ton mile for consolidation, having increased tonnage 37.5 per cent with performance of 27 per cent less coal per 100-ton mile.

PERFORMANCE ON BUTTE DIVISION

21 There are now two L-1 engines in transfer service between Butte and Woodville, where the ruling grade is 2.2 per cent. Their performance compares very favorably with that on the Cascade Division.

22 The L-2 engines have done good road service between Clancy and Woodville, on this division, replacing the F-8 engines where the tonnage was increased from 550 to 700 tons going west over a 2.2 per cent grade and from 1200 to 1650 going east. Owing, however, to another part of this branch having been washed out in the early summer, and track conditions not allowing for the handling of heavy trains east of Clancy, the Mallet engines between Clancy and Woodville were taken to other divisions until such time as the handling of heavy trains on this district would be warranted. The two Mallets in transfer service between Butte and Woodville, however, are still retained in service and are doing business formerly done by five consolidations.

GENERAL

23 The performance of Mallet engines above referred to by divisions shows their economy in fuel, also how they have increased the tonnage per train and thus cut down the cost of train service, as it

may be said that two trains with Mallet engines have replaced three trains hauled with consolidations.

24 The cost of maintenance one would naturally expect to be higher on the Mallet engines, but for the year ending June 30, 1908, the cost of repairs on the L-2 road engines was 10.47 cents, which we do not think at all excessive when it is considered that we have two pairs of cylinders and two engines but only one boiler to maintain. The cost of maintenance on consolidation engines in the same service is very seldom less than eight cents per mile.

25 Another feature which has been noticed from the start is that the Mallet engine is not at all hard on draw bars, principally because it is not possible in starting a train to slip both sets of drivers at the same time, so that the train is not jerked as it would be by a simple engine slipping and catching. When the Mallet engine does slip it is usually the high-pressure engine which slips first. This naturally builds up the receiver pressure and will cause the low-pressure engine to slip until the receiver pressure is worked down and the engines get down to equal work.

26 The tire-wear on these engines is very light and the flange-wear is also not excessive. In fact, Mallet engines have been put in on some districts where the flanges on consolidations would cut badly, and no appreciable flange wear has been noticed on the Mallets, no doubt on account of the short rigid wheel-base as well as the guiding pony truck. It is hardly necessary to state that the Mallet engine with its rigid wheel base of only 10 ft. and axle load not to exceed 50,000 is not hard on the track.

27 We still have in service two of the first L-1 Mallet engines, which have ever yet been in the shop for general overhauling, in fact, have never been off their wheels, but have been in continual service since October 1906, with the exception of light roundhouse repairs. The above performance would seem somewhat exaggerated, in fact the correctness of these statements has very naturally been questioned on several occasions by motive power officials not acquainted with the Mallet engines. Fortunately, however, some of them have had the privilege of investigating for themselves and I think it can be fairly stated that no one who has seen them in operation has been disappointed with their performance. While not wishing to say anything to retard the progress of the electric locomotive, we cannot help feeling that the introduction of the Mallet engine has set back the introduction of the electric locomotive for a great deal of hill service, as the Mallet performance has set a new figure for economical performance.

The Author It seems from the discussion that the only point of difference in opinion is whether a truck should be used or not. Its necessity cannot be proved by simply putting it there, but by omitting it, and the numerous cases of observation of engines properly designed with the omission of the truck prove conclusively that it is not required, and have in every way borne out the argument made in the paper for its omission. If a railroad, however, insists on having the truck and carries the responsibility for the service of the engine, it will of course have to be applied, but they would do better to leave it off, as it necessarily reduces the efficiency of the engine by its added resistance.

2 Mr. Vauclain said that European builders have applied the trucks on this type of engine, but so far as this is done it has generally been confined to engines with only two pairs of driving wheels in each set of engines, and where the curvatures have been so sharp that so short a wheel has been considered insufficient for the safe guiding of the engine, or in some cases where the builders and the roads ordering such engines have been in doubt as to their qualifications and have therefore added the trucks as a safeguard.

3 The oldest and most successful users and builders of this type of engine do not apply the truck, with an occasional exception of the two-axle class referred to above.

4 I heard with pleasure Mr. Vauclain's high tribute to Mr. Mallet, in which I join most heartily and hope that he may live to see his labor fully recognized, which is seldom the case with the prominent men who are in advance of their contemporaries. In regard to the cross-compound engine, however, Mr. Vauclain's statement that it has proved to be on a wrong principle will need some modification, so far as this continent is concerned. There are cross-compound locomotives in use which have shown for years a saving in fuel of from 25 to 40 per cent from the start to the present day. One of the greater trunk lines has for the last ten years built this type of engine exclusively, for its heaviest freight service, amounting to several hundred engines, and has found by frequent and thorough tests (the latest within the last few months) that performances are as good today as they were ten years ago, these engines hauling 45 per cent more load than the simple engines, to the same amount of fuel. They find them also considerably easier on repairs, using one set of tubes, to two sets on the corresponding simple engine, and they make 20 to 30 per cent more mileage between the repairs. These facts can be verified by anyone interested, by personal inspection, as they are

in service every day, and I have no doubt but that the officials of the road will give any desired information about them.

5 The continually increasing weight of the locomotive in general necessitated the compound to follow, and when 220 000-lb. engines were required, the size of the L. P. cylinder became too large for the limited clearances on most roads; the articulated type of engine described is simply an outgrowth or duplication thereof in all its essential features and advantages, with the adoption of the Mallet principle of articulation. With the expectation that the discussion would bring forth some particulars as to the performance of this engine, reference to it was purposely omitted from the text of the paper, and a few points of comparison with the 100-ton simple engines used in the same service may be of interest.

6 In pushing-work it took the place of two of these engines and did the work with ease. Special attempts were made to stall it by shutting off the front engine on the heaviest grade under full load, based on the capacity of three of the other engines, but it pushed both train and front engine to the top of the grade.

7 In road service it is rated to 2400 tons over a given division where two of the 100-ton simple engines are rated to 2200 tons (double headers), consuming practically the same amount of fuel as one of the simple engines. In order to find out the maximum power of the engine it was decided to make up a train of 2700 tons weight, but this did not stall the engine and another train of 2900 tons was tried with the same result; apparently no further attempt to stall her by overload has been made.

8 Another engine of this particular type and practically the same size has shown fully as good results in an entirely different kind of service, namely, pushing a snow-plow, where its extreme power at slow speed comes in most advantageously. Under these conditions it has successfully handled the snow-plow alone where the ordinary engine will stall and where it formerly required five 100-ton engines to do this work on various occasions.

9 One of the Erie engines, referred to by Mr. Cole, on one occasion started a train of four ordinary engine loads and the front engine on the grade, after having fixed a broken air hose.

10 These facts prove the statement in Par. 52 that, due to extreme power in running slow, the starting is affected without jerks and shocks, making it less destructive on cars and couplers than with much lighter ordinary engines that always have to jerk their trains into motion, utilizing whatever slack can be had in couplers.

No. 1221

TRAINING WORKMEN IN HABITS OF INDUSTRY AND COÖPERATION

By H. L. Gantt, Pawtucket, R. I.

Member of the Society

The widespread interest in the training of workmen which has been so marked for several years is due to the evident need for better methods of training than those now generally in vogue.

2 The one point in which these methods as a class seem to be lacking is that they do not lay enough stress on the fact that workmen must have industry as well as knowledge and skill.

3 *Habits of industry* are far more valuable than any kind of knowledge or skill, for with such habits as a basis, the problem of acquiring knowledge and skill is much simplified. Without industry, knowledge and skill are of little value, and sometimes a great detriment.

4 If workmen are systematically trained in habits of industry, it has been found possible, not only to train many of them to be efficient in whatever capacity they are needed, but to develop an effective system of coöperation between workmen and foremen.

This is not a theory, but the record of a fact.

5 It is too much to hope, however, that the methods about to be described will be adopted extensively in the near future, for the great majority of managers, whose success is based mainly on their personal ability, will hesitate before adopting what seems to them the slower and less forceful policy of studying problems and training workmen; but should they do so they will have absolutely no desire to return to their former methods.

6 The general policy of the past has been to drive, but the era of force must give way to that of knowledge, and the policy of the future will be to teach and to lead, to the advantage of all concerned. The vision of workmen in general eager to coöperate in carrying out the

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

results of scientific investigations must be dismissed as a dream of the millennium, but results so far accomplished indicate that nothing will do more to bring about that millennium than training workmen in habits of industry and coöperation. A study of the principles on which such training has been successfully established will convince the most skeptical that if they are carried out good results must follow. An outline of these principles has already been submitted to the Society in a paper entitled A Bonus System of Rewarding Labor.[1]

7 Under this system each man has his work assigned to him in the form of a task to be done by a prescribed method with definite appliances and to be completed within a certain time. The task is based on a detailed investigation by a trained expert of the best methods of doing the work; and the task setter, or his assistant, acts as an instructor to teach the workmen to do the work in the manner and time specified. If the work is done within the time allowed by the expert, and is up to the standard for quality, the workman receives extra compensation in addition to his day's pay. If it is not done in the time set, or is not up to the standard for quality, the workman receives his day's pay only.

8 This system, in connection with the other work of Mr. F. W. Taylor, so greatly increased the output and reduced the cost of the work in the large machine shop of the Bethlehem Steel Company that for the past seven years the writer has given a large portion of his time to the development of its possibilities. The results have far exceeded his expectations.

9 In his closing remarks on the above paper, the writer emphasized the value of the system as a means of training workmen, and the late Dr. Robert H. Thurston in his discussion of it was so optimistic as to the results it would produce on "workmen and foremen and employer alike" that the writer felt that his enthusiasm over a new and promising method had carried him, perhaps, a little too far. Results have fully justified Dr. Thurston's predictions, however, for today the method has been developed as a practical system of education and training for all, from the highest to the lowest. The fact, so repeatedly emphasized by Mr. Taylor, *that tasks should be set only as the result of a scientific investigation*, has proved of an educational

[1] A Bonus System of Rewarding Labor, December 1901, a system of task work with a bonus which had recently been introduced by the writer into the large machine shop of the Bethlehem Steel Company, as a part of the system of management being introduced into their works by Mr. F. W. Taylor.

value hardly to be over-estimated, for the scientific investigation of a process that has been developed without the assistance of science almost always reveals inconsistencies which it is possible to eliminate, thus perfecting the process and at the same time reducing its cost.

10 It is this scientific investigation that points to improvement in methods and educates owners and managers; but the average workman is interested only in his daily wage and has no special desire to learn improved methods. The results of our investigations are of little practical value, therefore, unless we can first teach our workmen how to use them, and then can induce them to do as they are taught.

PRACTICAL APPLICATION

11 For this purpose *an instructor, a task and a bonus* have been found most useful. People as a rule prefer to work at the speed and in the manner to which they have been accustomed, but are usually willing to work at any reasonable speed and in any reasonable manner, if sufficient inducement is offered for so doing, and if they are so trained as to be able to earn the reward. In carrying out this plan we try to find men who are already skilled and able to perform the task set. It frequently happens, however, that the number of such men is insufficient and it takes time to train the unskilled to a proper degree of efficiency; but with a bonus as an incentive, and a proper instructor, a very fair proportion of the unskilled finally succeed in performing a task that was at first entirely beyond them.

12 Unskilled workmen, who under these conditions have become skilled in one kind of work, readily learn another, and soon begin to realize that they can, in a measure at least, make up for their loss in not having learned a trade.

13 As they become more skilled, they form better habits of work, lose less time and become more reliable. Their health improves, and the improvement in their general appearance is very marked. This imprővement in health seems to be due to a more regular and active life, combined with a greater interest in their work, for it is a well known fact that *work in which we are interested and which holds our attention without any effort on our part, tires us much less than that we have to force ourselves to do.* The task with a reward for its accomplishment produces this interest and holds the attention, with the invariable results of more work, better work and better satisfied workers.

14 The Task and Bonus method of training not only furnishes the workman with the required knowledge, but by offering an induce-

ment to utilize that knowledge properly, trains him in proper habits of work.

HABITS OF WORK

15 In all work both *quantity* and *quality* must be considered, and our task method demands a maximum quantity, all of which must be up to the standard for quality. Workmen trained under this method acquire the habit of doing a large amount of work well, and disprove the oft-repeated fallacy that good work must be done slowly. As a matter of fact, our quickest workers almost always do the best work when following instructions. We set great store by the habit of working quickly, for no matter how much skill a workman may have, he will not attain the best success without quickness as well.

16 *Habits of work* in a mechanic are comparable with *habits of thought* in an engineer, and our industrial schools should make proper habits of work the basis on which to build their training in manual dexterity. The engineering school does not make engineers, but tries to furnish its graduates with an equipment that will enable them to utilize readily and rapidly their own experience and that of others. In the same manner, industrial training schools should equip their graduates with habits of industry that will make them as mechanics capable, and willing to do a large amount of good work.

17 As the writer sees it, one of the most valuable assets that the graduate of a technical college or an industrial school can have *is the habit of doing promptly and to the best of his ability the work set before him.* With this habit and reasonable intelligence he can make good progress. This habit is one of the first results of the Task and Bonus system, for it is a noticeable fact that task workers form habits of industry which they maintain even when on day's work with no bonus in sight.

18 In all schemes for technical or industrial education or training that the writer has seen, emphasis has been laid on the importance of knowing how. The writer wishes to add that ability and willingness to do are of at least equal importance. Many skilled workmen make their skill an excuse for slow work. Those that have not been trained to utilize efficiently what they have learned never attain the success that should be theirs.

19 Under our task system the workman is *taught how* and *trained to do* at the same time. *Knowing* and *doing* are thus closely associated in his mind, and it is our experience that the habit of doing efficiently

what is laid out for him becomes so fixed that he performs without hesitation tasks at which a man not trained to follow instructions would absolutely fail. This is exactly what should be expected and means nothing more than that in our industrial army the workman who has gained confidence in his superior follows his orders without hesitation, just as the private soldier follows the orders of his officer even though he does not see where they lead.

20 This is not a fanciful comparison, for I have known more than one case in which a workman expressed his doubts as to the possibility of doing a task, and on getting the reply that the task was all right, said, "If you say it can be done, I will do it."

21 Workers who have been unable to perform their tasks in the time set have frequently asked to have an instructor stand by them with a stop-watch to time the detail operations and show them just wherein they failed, with the result that they soon learned to earn their bonus regularly.

22 The first essential for a workman to become successful under our task system is to obey *orders*, and having acquired this habit he soon finds out that a skilled investigator can learn more about doing a piece of work than he knows "off hand." Having satisfied himself on this point, he goes to work at the tasks set him with the determination to earn his bonus, with the result, if he has the natural ability, that he soon becomes a rapid and skillful workman.

23 Learning to obey orders is often the hardest part of the workman's task, for a large percentage of men seem so constituted as to be apparently unable to do as they are told. As a rule, however, this is a feature of a certain stage of their development only, which under proper conditions they overcome at a later date. For instance, many very capable men who were impatient of restraint when they should have learned a trade, find themselves at the age of twenty-five or less in the class of unskilled workmen, although their ability would have enabled them to do well at almost any trade. It is this class of men, when they have come to realize the difference between a skilled workman and one not skilled, that furnishes us with many of our best task workers. Such men often see in our *instructor*, *task*, and *bonus* a chance to redeem some of their earlier errors, and by learning thoroughly how to do, and doing one thing after another, in the best way that can be devised, get a training in a short time, that does much to make up for the previous neglect of their opportunities.

BOSSES AS SERVANTS AND TEACHERS

24 In a shop operated on this system, where each workman has his task, one man whom we term a *gang boss* usually tends a group of workmen, supplying them with work and appliances and removing the work when finished. Such a man is paid a bonus for each workman who earns a bonus, and an extra bonus if all of his group earn their bonuses. The result is that so long as the workmen perform their tasks, though nominally their boss, he is really their servant, and becomes the boss only when a workman fails to perform his task. The loss of money to the gang boss in case a workman fails to earn his bonus is such that he constantly has his eye on the poor workman and helps him all he can. If, however, he finds that the workman is incapable of being taught, he uses his influence to have a better man put in his place.

25 In starting a shop on task work, an instructor who is capable of teaching each workman how to perform his task must be constantly on hand, and must as a rule teach one workman at a time. This instructor may be the man who has investigated the work and set the task, or he may simply be an instructor capable of following out the work of such an investigator, but he must be readily available as long as any of the workmen need his services, for we make it a rule not to ask a man to do anything in a certain manner and time unless we are prepared to show him how to do it as we specify.

TASK SETTING

26 A task must always be set for performing a definite operation in a specific manner, a minimum time being set for its accomplishment. As compensation, the workman is paid for the time set plus a percentage (usually 20 to 50) of that time, provided the work is done in the time allowed or less. If the time taken is more than the time allowed, the workman gets his day's pay only. The fact that in setting the task the manner of performing the operation is specified enables us to set another task for the same operation if we develop a better or quicker method.

27 If after having performed his task a workman wishes to suggest a quicker or better method for doing the same work, he is given an opportunity if possible to demonstrate his method. If the suggested method really proves to be quicker or better, it is adopted as the standard, and the workman is given a suitable reward. *No*

workman, however, is allowed to make suggestions until he has first done the work in the manner and time specified.

28 It is the duty of the investigator to develop methods and set tasks, and unless the methods developed by him are pretty generally a great deal better than those suggested by the workmen, he is not retained in the position. Working at tasks is pretty good training for task setting, and the writer has gotten more than one task setter from the ranks of task doers.

29 Inasmuch as, after a satisfactory method has been established, a large proportion of the work of the task setter is the study of the time in which operations can be performed, he is popularly known as the *Time Study* man. This term has lead to a misconception of his duties and has caused many honest people to claim that they were putting in our methods when they have put a stop watch in the hands of a bright clerk and told him to find out how quickly the best men were doing certain work. Unquestionably they have in many cases been able to set more accurate piece rates by this method than they had been able to set by the older methods, but they are still far from our ideal, in which the best expert available investigates the work, standardizes the appliances and methods and sets a task that involves utilizing them to their very best efficiency. While the stop watch is often used to establish a method, it is used to determine the time needed to do the work only when the standard methods and appliances are used efficiently. Stop watch observations on work done inefficiently or with ill-adapted appliances, or by poor methods, is absurd and serves only to bring into disrepute all work in which the stop watch is used. Moreover, such use of the stop watch justly excites the contempt and opposition of the workman.

30 To make real and permanent progress, the expert must be able to standardize appliances and methods and write up such instructions as will enable an intelligent workman to follow them. Such standards become permanent, and if the workman is paid a proper bonus for doing the work in the manner and time set, he not only helps maintain the standards, but soon begins to exert his influence to help the progress of standardization.

STANDARDIZATION

31 All work, and all knowledge, for that matter, may be divided into two classes: *expert* and *standard.* Expert knowledge may be described as that which has not been reduced to writing in such a

manner as to be generally available, or exists only in the minds of a few. By analogy, expert work is work, the methods of doing which either are known only to a few or have not been so clearly described as to enable a man familiar with that class of work to understand them.

32 On the other hand, standard methods are those that are generally used, or have been so clearly described and proved that a man familiar with that class of work can understand them and safely employ them.

33 The largest problem of our expert is to standardize expert methods and knowledge. When a method has been standardized, a task may be set, and by means of *an instructor and a bonus* a method of maintaining that standard permanently may be established. With increasing efficiency on the part of the workman the standard always has a tendency to become higher.

34 We have here the workman and the foreman using their efforts to maintain standards, for both fail to obtain a bonus if the standard is not maintained. This is so different from the case in which the standard is maintained only by the man in authority with a club that there can be no comparison.

35 From workmen trained under these methods, we get a good supply of instructors and foremen, and occasionally an investigator. From our investigators, who standardize our methods and appliances, we get our superintendents, and our system of management thus becomes self-perpetuating.

36 The superintendent who believes that the sovereign cure for all troubles is to go into the shop and raise a row, has no place under our methods, for when the task and bonus has been established, errors are far more frequent in the office than in the shop, and the man who is given to bluffing soon finds that his methods produce no effect on men that are following written instructions.

OBSTACLES

37 Among the obstacles to the introduction of this system is the fact that it forces everybody to do his duty. Many people in authority want a system that will force everybody else to do his duty, but will allow them to do as they please.

38 The Task and Bonus system when carried out properly is no respecter of persons, and the man who wishes to force the workman to do his task properly must see that the task is properly set and that

proper means are available for doing it. It is not only the workman's privilege, but his duty, to report whatever interferes with his earning his bonus, and the loss of bonus soon educates him to perform this duty no matter how disagreeable it is at first. We investigate every loss of bonus, and place the blame where it belongs. Sometimes we find it belongs pretty high up, for the man who has neglected his duty under one system of management is pretty apt to neglect it at first under another. He must either learn to perform his duty or yield his place, for the pressure from those who lose money by his neglect or incompetence is continuous and insistent.

39 This becomes evident as soon as the task and bonus gets fairly started, and the effect is that opposition to its extension develops on the part of all who are not sure of making good under it, or whose expert knowledge is such that they fear it will all soon be standardized. The opposition of such people, however, is bound to give way sooner or later, for the really capable man and the true expert welcome these methods as soon as they understand them.

HELPS

40 The fact that the task and the bonus enable us to utilize our knowledge and maintain our standards, and that the setting of tasks after a scientific investigation must necessarily not only increase our knowledge but standardize it, brings to our assistance the clearest thinkers and hardest workers in any organization. Our greatest help, however, comes from the workmen themselves. The most intelligent soon realize that we really mean to help them advance themselves, and the ambitious ones welcome the aid of our instructor to remove obstacles that have been in their way for perhaps years. As soon as one such man has earned his bonus for several days, there is usually another man ready to try the task, and unless there is a great lack of confidence on the part of the men in the management, the sentiment rapidly grows in favor of our task work.

DAY WORK AND PIECE WORK

41 As used by the writer, the Task and Bonus System of pay is really a combination of the best features of both day and piece-work. The workman is assured his day rate while being taught to perform his task, and as the bonus for its accomplishment is a percentage of the time allowed, the compensation when the task has been performed is a fixed quantity, and is thus really the equivalent of a piece rate.

Our method of payment then is piece-work for the skilled, and day-work for the unskilled, it being remembered that if there is only work enough for a few, it will always be given to the skilled. This acts as a powerful stimulus to the unskilled, and all who have any ambition try to get into the bonus class. This cannot be too clearly borne in mind, *for we have here all the advantages of day-work combined with those of piece-work without the disadvantage of either*, for the day-worker who has no ambition to become a bonus-worker usually of his own accord seeks work elsewhere, and our working force soon becomes composed of Bonus-Workers, and day-workers who are trying to become Bonus-Workers.

COÖPERATION

42 When 25 per cent of the workers in a plant are Bonus-Workers, they, with those who are striving to get into their class, control the sentiment, and a strong spirit of coöperation develops. This spirit of coöperation in living up to the standards set by the experts, which is the only way a bonus can be earned, benefits the employer by the production of

More work.
Better work.
Cheaper work.

It benefits the workmen by giving them

Better wages.
Increased skill.
Better habits of work.
More pleasure and pride in their work.

43 Not the least important of these results is the fact that the workmen take more pride in their work, for this of itself insures good work. As an instance of this pride, the writer has known girls working under the task system to form a society, admission to which was confined to those that could earn bonus on their work; the workers themselves thus putting a premium on industry and efficiency.

44 The fact that we get better work as well as quicker work seems inconceivable to some. The reasons are:

a Careful inspection, for no bonus is paid unless the work is up to the standard.

b Work done by a prescribed method, and always in the same way.

c Attention needed to do high-speed work, which keeps the mind of the worker on what he is doing and soon results in exceptional skill.

45 The development of skilled workmen by this method is sure and rapid, and wherever the method has been properly established, *the problem of securing satisfactory help has been solved.* During the past few years while there has been so much talk about the "growing inefficiency of labor," the writer has repeatedly proved the value of this method in increasing its efficiency, and the fact that the system works automatically, when once thoroughly established, puts the *possibility of training their own workmen within the reach of all manufacturers.*

TRAINING HELP A FUNCTION OF MANAGEMENT

46 Any system of management that did not make provision for obtaining proper materials to work with would be thought very lax. The day is not far distant when any management that does not make provision for training the workmen it needs will not be regarded as much better, for it is by this means only that a system of management can be made permanent.

47 To be satisfied to draw skilled workmen from the surplus of other plants means as a rule that second rate men only are wanted, and indicates a lack of appreciation of the value of well trained, capable men. The fact that few plants only have established methods of training workmen does not necessarily mean that the managers are satisfied with that condition, but rather that they know of no training system that can be satisfactorily operated in their plants; and as questions are sure to be asked about the method of introducing this system, a few words on that subject may not be amiss, it being borne in mind that *the changing of a system of management is a very serious matter, and cannot be done by a busy superintendent in his spare time.*

METHOD OF INTRODUCTION

48 In order to set tasks we must know beforehand what work is to be done and who is to do it. In order to pay a bonus, we must know after the work is done whether it was done exactly as specified. Hence our first care in starting to introduce this method is to provide means for assigning tasks to the workmen, and means for obtaining such a

complete set of returns as will show just what each man has done. When this much has been introduced, the output of a plant is always increased and the cost of manufacture reduced.

49 The next step is to separate such of the work as is standard, or can be readily made standard, from the more miscellaneous work, and set tasks for the standard work. Then we begin to standardize, and as fast as possible reduce the expert and increase the routine work. The effort to classify and standardize expert knowledge is most helpful to the experts themselves, and in a short time they begin to realize that they can use their knowledge far more efficiently than they ever dreamed.

50 As soon as work has been standardized, it can be intelligently planned and scheduled, each workman being given his specific task, for which he is paid a bonus when it is done in the manner and time specified. As bonus is paid only on the written statement of the inspector that the whole task has been properly done, failure to earn a bonus indicates that our plans have not been carried out.

51 An investigation of every case of lost bonus keeps the management closely in touch with the progress of the work, and as the workmen are ever ready to help disclose and remove the obstacles that prevent their earning their bonus, the managing problem is greatly simplified; for, as one of my co-workers has very aptly put it, "*the frictional lag due to the inertia of the workman is changed by the bonus into an acceleration.*"

52 With increase in the number of bonus workers, this force of acceleration increases, and not only does the careless worker, who by his bad work prevents some other from earning his bonus, fall into disfavor, but the foreman or superintendent who is lax in his duty finds his short-comings constantly brought before him by the man whose duty it is to investigate all cases of lost bonus.

MORAL TRAINING

53 The fact that under this system, everybody, high and low, is forced by his co-workers to do his duty, for some one else always suffers when he fails, acts as a strong moral tonic to the community, and many whose ideas of truth and honesty are vague find habits of truth and honesty forced upon them. This is the case with those in high authority as well as those in humble positions, and the man highest in authority finds that he also must conform to laws, if he wishes the proper coöperation of those under him.

DISCUSSION

Dr. Alex. C. Humphreys expressed the opinion that if this system were generally introduced throughout the United States, the resulting moral uplift would attract more attention than the increase of dividend-earning capacity. He said in part:

2 Mr. Gantt refers to the complaint as to the growing inefficiency of labor. The complaint is well founded, but certainly the responsibility cannot rest upon the working class alone. What has been done to meet the demand for trained workers in connection with the radical changes introduced into our industries since the days of the old apprentice system? Some well-directed and successful but isolated schemes have been inaugurated. Considerable attention is being paid to industrial education, and the methods of the author should receive careful attention in this connection as a substitute for the old apprentice system.

3 Mr. Gantt refers to a force not sufficiently recognized, which tends to encourage the worker to acquire greater efficiency and capacity as an earner, that of the pleasure and pride experienced by the performer in work well done. This is a force we cannot afford to ignore.

4 Our youths are not sufficiently taught the all-important lesson of obedience. We may well welcome any system which promises systematically to teach obedience to the employee while keeping the employer satisfactorily reminded that fair play is the price of loyal and efficient service.

5 Mr. Gantt advocates teaching the workmen at the same time *how* and *to do*. A more valuable lesson could not be taught in a country where the people aim to govern themselves.

Mr. H. V. R. Scheel[1] This paper contains many statements which must seem to the practical man and the engineer but expressions of what has been known for a long time, although perhaps not fully realized. Many of them are almost axiomatic. I think no one can doubt that such a system of management must result in the greater efficiency of workmen, individually and collectively, and in the greater coöperation of workmen, foremen and managers, with the attendant economies.

[1] The Brighton Mills, Passaic, N. J.

Discussions on this paper are here given in abstract only. They were published in full in the February 1909 issue of The Journal.

2 Under this system all the men are pushers to the limits set by the men responsible for quality. The gang bosses and foremen who were formerly drivers are now principally engaged in planning work and in handling extraordinary difficulties. Those of us who have seen these principles and methods in operation can testify to the correctness of the statements. We have seen men working no harder than before, but having been taught proper methods, accomplish results which make bonuses of 50 per cent on the former wages profitable for the owners. We have seen workmen, who without a trade have come to be considered and to consider themselves skilled workmen in a class as high as the trained mechanic. We have seen the removal of room mechanics, foremen, and even superintendents, when the workmen could no longer afford to permit the mistakes and neglects of these superiors to pass without protest.

MR. T. F. KELLY[1] Mr. Gantt's paper seems to me far too mild in its statements. My experience with the system for the last two years has been that every man in the plant, whatever his authority, has a specific job, and will get into trouble if he lays down on it. Under this system every employee becomes also an inspector, and for fear of losing his bonus protests vigorously against accepting material on which he has to work, unless both the material and former work are up to the standard for quality. It takes only a few touches on his pocketbook to make Mr. Jones a first-class critic on Mr. Brown; in other words, 300 employees means 300 inspectors.

2 The machinery must also be in first-class condition for operators to make their bonus. The "good enough" machinist cannot live under this system, as he not only loses his own bonus, but causes the gang-boss and the operator to lose theirs. These all act like a tonic on our Mr. Machinist, to do a good, quick job, or he soon finds out he is not the man.

3 The introduction of the bonus system is followed by a new feeling of pride, and the threat to move workmen from machines on bonus to machines not on bonus is more effective than the threat to lay off or discharge.

4 The operators in our factory formerly measured their work to the clock; they now measure it to the task, and former shirkers are now among the most zealous of our employees.

DR. RUDOLF ROESLER[1] called attention to the fact that in Europe

[1] The Brighton Mills, Passaic, N. J.

the greatest interest exists in the new ideas of economical organization and management in workshops, among which are those of Mr. Fred. W. Taylor. One important consideration is the belief that the new principles can help to weaken the *casus belli* for the struggle between capital and labor, to weaken the reasons for existence in continental Europe of Social Democrats and in America of Labor Unions, and to unite the workman and his employer by making their interests common. He mentioned the following to illustrate the general interest:

2 Der Verein deutscher Ingenieure has resolved by almost unanimous vote of its branch societies to give a place to such questions in the Zeitschrift, and a number of scientific practical men have promised to support this movement.

3 The tool machine works of Mr. Ludwig Loewe in Berlin is considered one of the best organized and managed factories of this kind. As much as is compatible with the German conditions this concern has been the first to take up the ideas which Mr. Gantt has explained. The book concerning the organization of the Ludwig Loewe Company by Mr. Lilienthal has proved the general interest by its wide circulation.

4 At the Technische Hochschule of Berlin in Charlottenburg there is a chair of Organization and Management, founded, I think, in 1904, where these principles are taught. According to the rules formed by the Prussian government and the Technische Hochschule, graduates in mechanical engineering, and I think also in civil and electrical engineering, must take examinations in those two subjects. Thus every year as many as 1000 engineers go into the world with a knowledge of these principles.

Mr. H. K. Hathaway The writer believes it possible under the Taylor system to turn out a first-class mechanic in about one-half the time taken under the old system of apprenticeship; an opinion borne out by results in a machine shop operated under the Taylor system with which he is connected. In this shop a number of young men, who came to us without previous experience at the machine trade, within a year and a half reached a point where they were capable of turning out work of excellent quality on any of the machines in the shop, and doing it in the time set by the planning department.

2 One thing to which the writer hopes Mr. Gantt's paper may lead is the adoption by the trades schools of the methods advocated. Most trades schools pay very little attention, if any, to the time taken by their students for performing the various exercises or tasks forming their course. Too often they have not nearly enough instructors, and

boys waste a great deal of time trying to figure out how to do the various jobs; furthermore they have no conception of the feeds and speeds and depth of cuts that should be used in doing work in machine tools.

3 An instruction card prepared for each piece of work, showing the manner in which it should be set in the machine and explaining the various steps of the operation in their proper sequence and the tools to be used, would not only make it easier for the instructor but would enable the student to learn at once the best method, instead of using a method of his own with no foundation in experience. If proper instructions and tools were furnished, and the machine and belts kept in good working order, a definite time could be placed on the job, and the student made to acquire habits of industry.

4 The greatest value of the system of training outlined in Mr. Gantt's paper lies in the fact that in busy times, when skilled workmen are unavailable, it is possible to train inexperienced men, who are intelligent and ambitious, to turn out good work rapidly. The writer has seen an absolutely green man trained under this system so that in less than a month he was capable of turning out work on a drill-press as satisfactorily as an old hand; of course during this time the "gang-boss," "speed-boss," and "inspector" were almost constantly with him helping and instructing him. After a workman has learned to run a drill-press successfully, he can be trained in about the same time to run a milling machine, lathe or planer.

5 One of the best examples of the efficiency of this system of training workmen, is the results achieved with young college students taken on after their freshman or junior year, in the shops with which the writer is connected. After their year in the shop they return to college and complete their course. One object of this plan is to train them in habits of industry, and this object is most successfully accomplished.

6 That the Taylor system is a system whose success is due to teaching and helping the workman, should be brought out more prominently. In the first place, his proper tools, in first class condition, are provided, and his machine and belts are kept in good repair. Secondly, the gang-boss, must show him how to set his work up quickly and in the best way, and not only tell him, but demonstrate

it. The inspector must not only detect defects in his work, but must explain, when the workman starts on a job, the drawings, the degree of accuracy, and the kind of finish required. The speed-boss instructs him in the actual operation of his machine, and the setting of his tools, feeds, speeds and depth of cuts, and is prepared to help him if necessary by actual demonstration.

7 Under this system a workman can turn out from two to four times as much work, as his efforts are not largely consumed in finding out what he is to do, devising ways to do it and struggling against discouraging adverse conditions over which he has no control.

Mr. Charles Piez The bonus system of rewarding labor, which Mr. Gantt describes in his paper, can hardly in itself be considered a system of instruction, and is, in fact, no more an instrument to this end than any of the well known schemes of compensating workmen. It is through the methods Mr. Gantt employs that his work becomes a most effective means for training workmen in habits of industry and coöperation. What appeals to me most in Mr. Gantt's presentation is its distinctly human tone; the spirit of helpfulness toward the worker which it evinces.

2 Mr. Gantt recognizes the fact that reörganization often means only a change of mental attitude, and that it can, therefore, be best accomplished by persuasion and example. Then, too, while establishing fixed methods of performing tasks, he allows ample opportunity for initiative on the part of the worker; in fact, he stimulates and directs it.

3 In these days when systematizing of industrial establishments has become a recognized specialty in the mechanical world, a few thoughts suggested by Mr. Gantt's paper may not be amiss.

4 There is abroad today a great deal of what might be termed system idolatry, which manifests itself in the belief that system produces output, when, as a matter of fact, it simply indicates the lines along which maximum output can be attained; and because of this erroneous conception, the system assumes the rigidity of a creed, and the various printed forms of which it makes use are invested with a sanctity that is intended to place them beyond the reach of suggestion or criticism, whereas they are frequently modified without any departure being made from the underlying principles.

5 The adaptability of an already existing organization, from which the material for carrying out a system must be drawn, the peculiarities of the product, and the demands of the customer must

be given full consideration. If the system is considered the important thing, and organization, product and customer must bend to its lines, is it any wonder that attempts at systematizing a plant may fail to result in the full economies promised? And they fail, not because the system is inherently wrong, but because of the fanaticism of the enthusiast applying it. Tact and good judgment must be supplied by the introducer or receiver of the system.

6 I am a firm believer in the efficacy of shop system, for in its essence it implies the production of work along lines of least resistance and greatest economy. But direct lines are not always the lines of least resistance, particularly when they run counter to peculiarities of ability or temperament in an otherwise efficient organization. It seems unnecessary to compel an organization to conform to a system chart, because it is much simpler and more effective to make the chart conform to the abilities of the individuals composing the organization.

7 The first step, even in the mildest form of reörganization, is a partial disruption of the existing organization, and great care and tact must be exercised, lest in the rebuilding, discontent and discord creep in. The line between profit and loss in most establishments is so fine that even a single element of discord can destroy that intangible, profit-making quality, known as team spirit. It is on this account that my interest lies, not so much in this system or that, as in the personality and methods of the men applying it.

Mr. C. N. Lauer It has been the writer's experience that the best results, after a definite plan has been laid out, are obtained if the spirit of coöperation can be engendered. The manager who depends entirely upon his own ability to drive his employees is bound to fail by just so much as the employee holds in reserve against contingencies which he feels may arise through the whim of the manager. It is the spirit of helpfulness which runs all through Mr. Gantt's paper that especially prompts the writer to pronounce it well worthy of serious consideration.

Mr. Lewis Sanders[1] cited individual cases where study of the methods of doing a piece of work resulted in a marked reduction of time and this often in classes of work where saving would not be expected; for example, in reading thermometers. He recalled a test where there were twelve thermometers to be read once every two minutes and the observer never succeeded in reading more than

[1] With General Engineering Co., New York.

eight and was on the go continually. He took the observer's records and tried to see what could be done. The first few readings he got no more than did the observer, but by studying just where to stand so as to get the light unobstructed and not be obliged to shift his position, he was in a short time able to read all the thermometers in 1 min. 50 sec. He further said:

2 At Schenectady I have seen a 2000-kw. vertical turbine used for experimental work, completely dismantled and a new set of wheels put in, and the machine re-erected and running within 24 hours from the time steam had been shut off. This was done by two machinists and a cranesman. I am informed that the men have now become so expert that they do it in twelve hours.

3 A factory in which I am interested bid on turning out a certain small piece of work in quantity. We made a detail study of every fractional operation involved and made our bid on our estimates. It was so low that it was returned to us with a request to revise it, as the customer was sure it was less than the labor and material cost. The customer had already made 80 000 pieces. We had sufficient confidence in our figures to insist on their acceptance as they stood. The customer was never convinced that we did not lose money although we made 44 per cent profit. But he had never been compelled to analyze every single step in turning out the goods, to find out what the cost should be. He had merely made some and knew what they did cost. We should probably have done the same if we had not been compelled to make the analysis. One of the operations on this job was the setting of the piece in a jig, the man who was put on it taking regularly 1 min. 40 sec. After a study of the exact motions required to pick the piece up and set it accurately we showed the same man how to do it in 20 sec.

4 In speeding up a shop a distinction should be sharply drawn between work done at high speed and work done in a hurry; the first will give perfect goods because the speed is attained by elimination of all the unnecessary motions, the latter bad work because it is a speeding up of all the operations, necessary and unnecessary. High speed work also involves less labor on the part of the employee, than slow speed work. The man who could read only eight thermometers was going all the time; but there was 10 sec. to rest when reading twelve.

5 There is no question but that it requires a considerable knowledge of men to introduce successfully such a system as Mr. Gantt advocates; the theory might be followed out exactly, and the whole system be wrecked by lack of common sense and a little knowledge

of human nature. One obstacle to securing high-speed work is the suspicion of the men that rates will be cut if they show that they are able to earn more than a certain sum per week; and they have been given ample grounds for this. On this account the time determined by expert analysis is the only reliable standard as the men will go as slow as they dare.

6 While quite true that there is a maximum wage that we can afford to pay, it is also true that an agreement with a workman should be lived up to as strictly as a contract with a customer. We have to fulfil our contracts even if we have made an error and find them unprofitable; the same should be done with the workman when we find that we have set too easy a task. Tasks set should therefore hold for a definite period, say a year, unless a change in method is made. Before the task is set it should be the subject of accurate investigation to determine the minimum time possible. If the task set proves too severe it must be corrected at once, as the workman is not a party to the setting of the tasks.

MR. J. C. JURGENSEN Mr. Gantt's methods are based on a concrete knowledge of the human element, which is sure to result in fair dealing to both men and employers.

2 I feel justified in speaking on this subject, since for more than five years I have used a sort of apprentice system for producing reliable and efficient help for the operation of power plants. Although the conditions in an engine room differ very widely from those in a shop or factory, yet the same principle holds, that men who can make good must be trained. When this is once fully realized, every shop and plant owner will look more favorably at the idea of maintaining a training course for his men, as a part of his routine business.

3 The leader must make his men understand that it is one of two things: advance, or make room for a better man; and inducements for compelling a man to see it in that light, such as a nine-hour work-day, reasonable wages, provision for advancement, etc., must be present. It is a question of setting right both the employer's mind and the workman's job, and the Golden Rule must be applied before success is possible. As a general rule, it is necessary to hold out material inducements for acquiring better skill and efficiency. With some, but not with many, ambition and the right temperament will bring this about.

4 To secure safety and economy in an engine room, the men

must become willing workers and must take pride in the engineer's department. To reach this point, a man must feel that his position is secure, with a good record, and that advancement and benefit are certain to follow; on the other hand, that a continued bad record will bring discharge. Advancement according to seniority is all right with a qualifying clause—instead of giving preference to the oldest man in the service, make it the *oldest best man.*

5 If opportunity for material advancement is not present when a man has gained his apprenticeship or has reached the highest station possible, we find a new job for him. This is not so hard as it would seem, since other plants are generally willing to take him, and this provides a further chance for advancing the members of the department. Success along this line cannot be had without coöperation, and this we obtain mainly through a system of rules, and an organized school of instruction in which the chief engineer is the chief instructor, and each man in charge of a section, an assistant instructor in that section of work.

6 In the fire room, the men are paid a good salary for producing one boiler horse power for a certain amount of coal containing a certain per cent of ash: 10 per cent of the value saved over this standard goes to the fireman as a cash bonus each month; on the other hand whatever is lost, according to the standard, is deducted from that man's monthly saving. The head fireman receives a sum equal to half the bonus of each fireman—he is therefore vitally interested in having each man do as well as possible. If a fireman cannot make a bonus, his job is not secure.

7 The coal passers also receive, divided equally between the three shifts, a sum equal to half the bonus of each fireman—they are therefore much interested to see that all the firemen "make good," and yet there is no motive for collusion between the men. It is soon shown that the best policy for both the company and the men is a continued effort to make the bonus as large as possible. All the men in the department are thus taught to follow both daily and monthly standards in work and expenses.

8 To illustrate the scope of our Engineering Department Training School, which is a part of the Men's Relief and Educational Society, I will mention the objects of the Society, as stated in our by-laws:

Section 1 The objects of this Society shall be the raising of funds to provide a weekly relief income to members in good standing during illness or accident and such other relief as may be deemed advisable, and to assist in defraying burial expenses of a deceased member; also, to defray the expenses incurred in carrying on the training course which constitutes the Educational Branch of this Society.

Section 2 As a further relief, members who are in good standing may apply to the Society for loans, not to exceed 10 days pay of such members' monthly salary. Loan to be paid back in four successive and equal monthly installments, plus one cent per dollar per month, on amount due to the Society.

Section 3 The object of the educational course is to give such practical instruction and example as will further a spirit of manhood and induce the members of the Department to become self-reliant, observing and manly men. Training such men to become safe and conscientious workmen, worthy to receive the Company's Certificate of Merit for two years service.

Section 4 To ambitious holders of the Certificate of Merit, the training course will endeavor to supply the technical information most needed to make such workmen qualify as safe and efficient Operating Steam Engineers, worthy to receive the Company's Operating Steam Engineers' Apprenticeship Certificate for five years service.

Mr. Willis E. Hall raised the question as to why the task system of itself promises better results than piece-work. The term piece work is here intended in its broadest sense, to include all methods of compensation based on the amount of work done. It therefore includes such modifications designated as the bonus, premium, differential, profit-sharing, gain-sharing, etc. On the contrary the task system seems to eliminate day work only partially. For instance if the employee can not do the work within the time set for the task he receives no additional compensation and his day rate prevails. Assume, then, that the delinquent is replaced by one who can meet the task (and the task system has no monopoly of procedure in this respect, as the same degree of vigilance should be exercised with any system), until the plant has, finally, the very best set of men obtainable; and further that the men are instructed to the highest degree of efficiency (and here again the task method does not possess exclusive privileges). It is unnecessary to substantiate the statement that there will still be a vast difference between the most efficient and the least efficient of the force.

2 How then will you set the task? If it is set so that but a limited number, say 65 per cent can meet it, then the remaining 35 per cent must be eliminated from the rewards. The answer may be that this remaining 35 per cent will have the day rate adjusted according to their relative efficiency. To that extent the task is not better than the day-rate system. The other alternative is to set a task that the least efficient can meet, but neither is this fair to the employer nor would it promote efficiency in the employee. In short, with the task system proposed men must be practically equal in efficiency or it is only a partial eliminator of day work. Unavoidable rul-

ing conditions due to the unequal efficiency of men, instructed or otherwise, seem to prevent this from being other than a partial day-work system for producers. However, this is of minor importance in comparison with the other feature in Mr. Gantt's paper, the use of an instructor.

3 For greater productive efficiency, to the mutual benefit of employer and employee, the use of an instructor, combined with the intelligent piece-work price-setting described by Mr. Taylor (Vol. 14 of Transactions), seems full of promise. That the use of an instructor establishes "habits of industry" of which Mr. Gantt speaks, must not be misconstrued. There is quite a difference, with apology for the terms, between the "sweating process" and "farming the work;" both extremes are equally demoralizing. It is the medium position that is always most difficult to assume at first and the one that is most desirable. It is only by accident it is reached by the haphazard method of establishing piece-work rates at present so generally used, which usually results in one or the other of the two extremes, until the price, after many adjustments, is about where it should have been originally. Intelligent setting of piece-work rates or task limits with an instructor eliminate the more flagrant abuses of the system. In concluding he said in part:

4 A feature that should be made a part of the duties of the instructing and piece-setting department is that of time-checking, where either the piece-work or the task system is in vogue. For instance a producer spends actually 7½ hours on piece-work but has been in the shop some 10 hours in all. There is a loss "for good and all" of 2½ hours of productive time. In other words, what percentage of the producer's time is lost in this way, and why? Is it not safe to say that most of the establishments using piece-work ignore this point? In one instance where the tonnage of a plant was increased more than 90 per cent over what was formerly, at least, normal tonnage, about 30 per cent of this increase was due to the elimination of the annoying and demoralizing delay known as "waiting for work." And yet this delay was previously not over-conspicuous and its magnitude was not appreciated until it was eliminated. There was practically no rate-cutting.

Mr. Harrington Emerson I once asked Mr. Gantt what happened in case one of the workmen whose time he had set managed to do the task in less than the time prescribed. Mr. Gantt very cheerfully answered that in that case the workman would take his place and

he would have to take the workman's place. Since then my assistants and I have had to standardize over 100 000 different jobs, and have never been able to realize the accuracy which Mr. Gantt claimed. The reason probably is that Mr. Gantt's work lay along standard lines, working on standard material, in standard manner, while my work lay in repair shops, working on unstandardized material, unstandardized jobs and unstandardized conditions. In such matters as locomotive repairs we were able to predetermine, within four per cent, both time and cost of the aggregate, but not to strike the exact time of the jobs that entered into repairs.

2 Now Mr. Gantt comes forward with this paper, and we have a chorus of assent to the statement that by training the workmen the problem is solved. My experience has been that this is not at all the end of it. It is easy to train the workman in habits of industry and coöperation, but when you have provided a method for training him, you have not touched the real problem. What I would like from Mr. Gantt is a paper on training managers in habits of logical thought and coöperation.

3 To illustrate the enormous practical and economical importance of this question, I quote the figures on locomotive repairs per mile on four out of the five trunk lines running west from New York. This is not a very accurate standard, but nevertheless it is used. On one of these, 44 mills maintain the power in first-class operating condition; on the second road, the cost is 7 cents; on the third road 12 cents; and on the fourth road over 16 cents. This fourth road has a mileage of 30 000 000 miles, and the cost is over 12 cents more than on the first road, making the amount of money lost per year $3 600 000. Yet this road has good grades out of New York, while the road that shows the lowest cost has the heaviest grades.

4 The trouble does not lie with the workmen. No doubt the workmen are wasteful, no doubt they have not been fully trained, but the trouble lies absolutely in the managements which have not yet awakened to their problems, and accept enormous wastes as inevitable.

Mr. Milton P. Higgins When this marvelous proposition was first advanced by Mr. Taylor, I looked upon it as a departure of very great promise, and I have not been disappointed in its results. In addition to considerations of shop discipline, efficiency, output, profits, and the well-being of the workmen, as a means of education, I believe it is equally important and efficient.

2 In place of the old system of apprenticeship we have now the training school, the apprentice school, the trade school; but the shop which trains its apprentices to the best advantage to meet modern requirements has within it a trade school. That is, the task, intelligent analysis of methods, and the bonus, are most effective elements of disciplinary education, not only important as developing skill and efficiency, but developing intellectual capacity and manly character. I believe that the training given in the shop through these methods may be as good as any line of mental and intellectual discipline that can be offered in a university.

3 In regard to the suggestion Mr. Emerson made, of training managers and superintendents, if we wisely and faithfully train the skilled workmen, are we not training the managers? If we want large trees in the forests, had we not better give our attention to the nursery?

Prof. Wm. Kent At the meeting in Detroit in 1895, when Mr. Taylor made an address upon a subject similar to this, he met with unanimous dissent from the older men; and I got up in that meeting and said that a man over fifty years of age could not be expected to appreciate Mr. Taylor's paper, and that the revolution in the industry that was to follow from Mr. Taylor's plan was to come from the work of such younger men as Mr. Gantt. I am glad to see that Mr. Gantt has verified my expectations.

2 The hopeful thing about this paper, which I regard as the most important that has ever appeared in the Transactions of the Society, is that it is in harmony with humanitarian ideas. I am glad to see this sentiment throughout the Society. There would be more of it if more of our mechanical engineers would go into shops, instead of into power plants and drafting offices.

3 The trouble with the managers has been mental inertia. Men of great brain power, enterprising, and in important positions, will not take half an hour, which is worth perhaps $50 to them, to think on a problem which might end in saving them $10 000 a year in the shop. That kind of manager, however, is rapidly being displaced and the younger men, who are willing to do some thinking along the lines of Mr. Gantt's paper, are coming forward to take their places.

Mr. James M. Dodge I have been asked whether the concern that I am connected with has abandoned the Taylor system, and I desire to be recorded as saying that we have not abandoned it; on

the contrary, it is working to our complete satisfaction in Philadelphia and Chicago, and we are introducing it as well in our Indianapolis plant. Mr. Taylor told us five years ago that the introduction of his system would be of great saving to us in normal times, but that its greatest value would be shown during periods of commercial depression, and I am very glad to be able to say that Mr. Taylor was absolutely correct in his statement. Our business during the past year (1908) would have shown a deficit had we been working under the methods we thought excellent prior to our adoption of the Taylor system, which carried us through this twelve months during which our business was curtailed about one-half, with a moderate profit showing on the right side of our annual statement.

(Question by Mr. Frank A. Haughton as to the difficulty in holding together their organization of experts through this period of depression.)

2 I am glad that question was asked, because we did not experience the difficulty suggested. Under the Taylor system the functions of the different members of our organization are so clearly defined, that we were able very promptly to reduce our expenses by demoting a number of our workers, thus avoiding the demoralization of the sub-divisions of our organization. We took the men into our confidence and explained to them the exact situation, and they cheerfully accepted it; in fact we had no trouble at all. Our organization was and is intact, and, as Mr. Taylor told us, can be readily expanded to any desired extent as soon as increased orders make it necessary.

The Author A system of management may be defined as a means of causing men to coöperate with each other for a common end. If this coöperation is maintained by force, the system is in a state of unstable equilibrium, and will go to pieces if the strong hand is removed. Coöperation in which the bond is mutual interest in the success of work done by intelligent and honest methods produces a state of equilibrium which is stable and needs no outside support.

2 Until within a few years the mechanic was necessarily the source and conserver of industrial knowledge, and on him rested, therefore, the responsibility for training workmen. With the advent of the scientifically educated engineer, capable of substituting a scientific solution of problems for the empirical solution of the mechanic, the responsibility of training workers naturally shifts to his shoulders. If he accepts this responsibility, and bases training on the results of scientific investigation, the efficiency of the workman can be so greatly

increased that the manufacturer can afford to give those that take advantage of this training such compensation as will secure their hearty and continuous coöperation, thus making the first permanent advance toward the solution of the labor problem.

3 This was first demonstrated by Mr. Fred. W. Taylor, and the whole industrial community has already been profoundly influenced by his work.

4 It was well said yesterday that the work of the engineer has been less appreciated than that of any other learned profession, and a broader recognition of his work will have a marked influence on our civilization.

5 He is carrying forward under the direction of science the work that was begun by the mechanic who first learned to chip flint or make a fire, and it is he alone who can lead the mechanic of today to a better understanding of his problems, and the capitalist to a better appreciation of their solution.

No. 1222.

SALT MANUFACTURE

MECHANICAL METHODS AND ENGINEERING FEATURES OF LARGE SALT PLANTS

By George B. Willcox, Saginaw, Mich.

Member of the Society

The literature readily available on the subject of salt manufacture is not extensive, and is for the most part contained in volumes devoted principally to other subjects, such for instance as geological reports of the various States, besides a number of pamphlets that have appeared from time to time as the result of special investigations of specific problems relating to salt manufacture. These pamphlets and reports have in general treated the question from the standpoint of the chemist or the geologist rather than that of the mechanical engineer.

2 It is the purpose of this paper to note from a mechanical engineer's point of view a few of the more recent developments in the mechanical methods and appliances of some of the large salt plants. Reference will be made solely to plants operated by what is known as the steam grainer system, as distinguished from the vacuum pan system, and the solar or open air system.

GENERAL ARRANGEMENT OF A GRAINER PLANT

3 Since the nature of the brine obtainable determines the treatment necessary before the actual process of evaporation begins, the arrangement of the component parts of a grainer plant for the manufacture of salt varies in different localities. The majority

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

in America are arranged somewhat as in the map (Fig. 1) of a salt plant having ten grainers, with a total producing capacity of approximately 1000 bbl., or about 140 tons per 24 hours. It may be of interest to note that this plant operates by exhaust steam of less than 2 lb. pressure. It operates 24 hr. a day. Six men are employed at the plant, four on the day shift and two at night. These men do all the work, including pumping, liming and settling the brine, making the salt and delivering it on the storehouse floor. Fig. 2 is a bird's-eye view of such a plant.

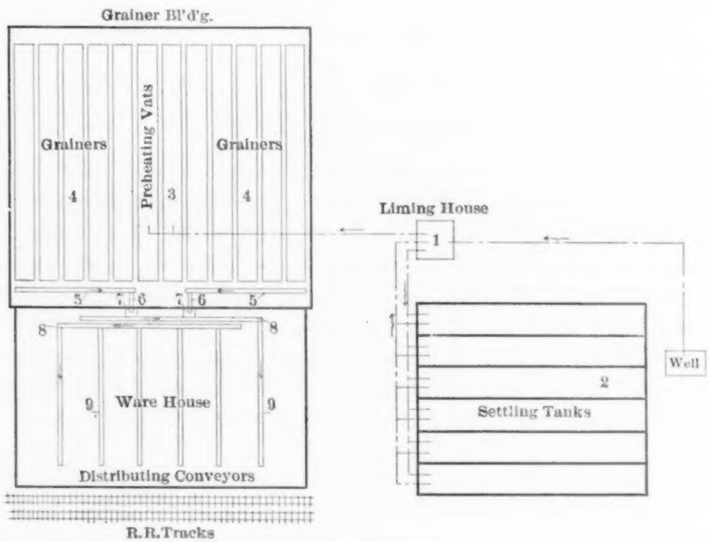

Fig. 1 Map of Salt Plant

4 In general, the mode of operation is as follows: The brine is first pumped from the wells into a liming house (1), where it is mixed with a suitable proportion of slaked lime to precipitate the iron contained in it. After liming, the brine is pumped into settling tanks (2), where it remains until perfectly clear, usually 24 to 48 hr. or longer being required for the settling. From the settler it is drawn into large pre-heating vats (3) located in the grainer building, and there heated by means of submerged steam pipes. From the pre-heating vats the brine is drawn into the grainers (4), where it is evaporated by steam pipes submerged in the brine.

5 The salt produced by each grainer is moved along the grainer bottom and up an incline at its front end by an automatic "salt raker" and then dropped into a collecting conveyor (5). Thence it is carried by a conveyor (6) to the foot of a bucket elevator (7) that hoists it to the top of the storehouse. The elevator discharges at its top on either of a pair of conveyors (8) running in opposite directions and located just beneath the storehouse roof. From the conveyor (8) the salt is delivered to any one of a series of distributing conveyors

FIG. 2 BIRD'S-EYE VIEW OF SALT PLANT

(9) and dropped therefrom to the warehouse floor. The height of the distributing conveyors (9) and their distance apart is such that nearly the entire storage space of the warehouse can be filled with salt without shoveling.

SALT GRAINERS OF REINFORCED CONCRETE

6 In years gone by white pine was available for the construction of grainers and was almost universally used. It withstood the action of the hot brine admirably, was easily worked, did not shrink to any great extent as do nearly all other woods when exposed to hot brine,

and when properly calked with oakum was capable of giving good service for five years or more.

7 With the decline of the white pine supply, however, came a demand for some material other than wood from which to build grainers and attention was directed toward reinforced concrete.

8 The ordinary salt grainer is a pan usually 150 ft. long by 12 ft. wide by 2 ft. deep equipped with steam pipes. This pan is often called upon to undergo considerable changes in temperature, varying in extreme cases from 40 to 185 deg. fahr. and back again to 40 deg. It must of course remain brine-tight under all conditions.

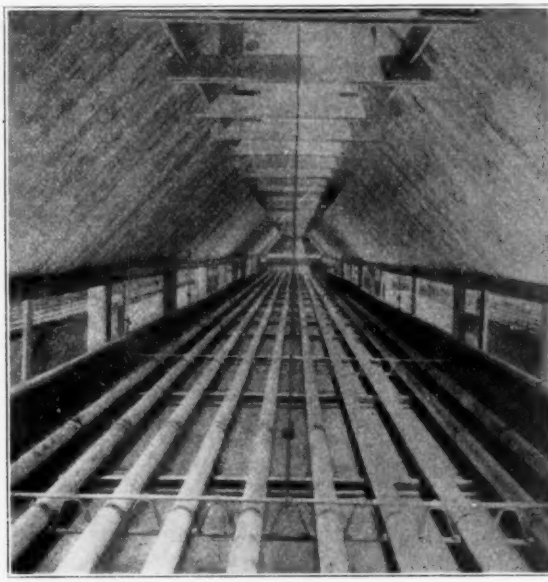

Fig. 3 Monolithic Concrete Grainers

9 A method of construction that has proven satisfactory to stand the strain incident to such temperature variation is shown in Fig. 3. These grainers are monolithic, no expansion joints being used. They rest on a hard flat bed of rammed sand, giving a uniform support throughout the grainer bottom and reducing radiation loss.

10 When the brine is warmed from say 40 to about 180 deg. fahr., the concrete grainer expands approximately $1\frac{1}{2}$ in. in its length of 150 ft. All the grainers do not expand alike, however. When cooled

they contract to their original position. After nearly 3 years of constant use they are in first-class condition and brine-tight. Concrete is an excellent material for reducing radiation loss; in fact these grainers continue to make salt for three or four days after the steam is shut off.

DETAILS OF CONCRETE GRAINER CONSTRUCTION

11 Their construction is shown in detail in Fig. 4. The grainer walls are 6 in. thick and the bottom 4 in. The reinforcing bars are ¼ in. corrugated. Transverse the grainer the bars are laid 4 in. on centers. Alternate transverse bars extend down one wall, bend at the

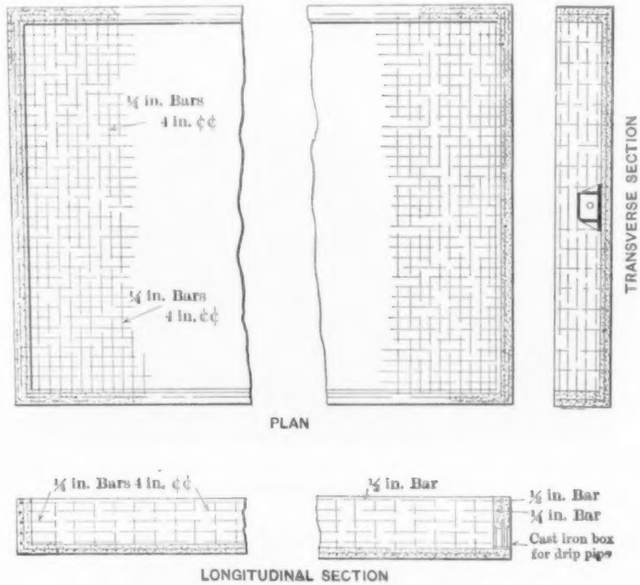

FIG. 4 DETAILS OF GRAINER CONSTRUCTION

juncture of the wall and bottom, and run imbedded in the bottom to the base of the opposite wall. The longitudinal bars are also spaced 4 in. apart and at the ends of the grainer are bent up to form vertical reinforcement. The side and end walls have horizontal bars spaced 4 in. apart, and near the top edges of the walls is a ½ in. bar extending entirely around the grainer. Fig. 5 shows a grainer under construction.

12 The concrete used was a mixture of Portland cement, sand and gravel. The lower 2 in. of the grainer bottom was in the proportion of one to six, being 1 cu. ft. of cement to 2 cu. ft. of sand and 4 cu. ft. of gravel. The top 2 in. was 1 cu. ft. of cement to 3 cu. ft. of sand. The side walls were 1 cu. ft. of cement to 1 cu. ft. of sand and 1 cu. ft. of gravel. All the concrete was mixed very wet, so that it would run off a shovel and was poured into place. No water-proofing method was used in the construction of the grainers other than a gentle spudding of the side walls next to the forms with a thin bladed spud immediately after the concrete was poured. The bottom was leveled by striking off with a strike

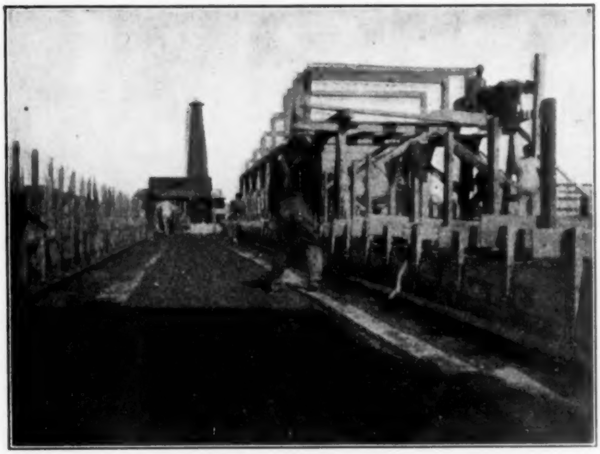

FIG. 5 LAYING CONCRETE IN GRAINER BOTTOM AND WALLS

guided at its ends on rails spiked to the sides of the wall forms, and the surface was finished by troweling. Openings for drip pipes, etc., were made by embedding cast iron boxes provided with anchor wings into the concrete, Fig. 4, and fitting these boxes with glands and packing boxes.

13 After becoming accustomed to the work by building two or three grainers, a crew of twelve men can place the reinforcing bars and pour the concrete for a grainer of the size given in 9 hr.

14 The first few grainers at this plant were built in the open air, and as the sun was very hot they were covered by a canvas tent, shown in Fig. 6, which was later dispensed with when the building

construction was far enough along to protect the fresh concrete. The same grainer with the forms removed is shown in Fig. 7.

15 The life of these reinforced concrete grainers has not been determined, but after three years' use the indications are that they are good for many years. If the hot brine shows a deleterious effect on the concrete after iong use, the side walls will be lined with flat slabs of vitrified tile, but the necessity has not arisen up to this time.

RAKERS

16 In the process of evaporation, the salt accumulates on the grainer bottom and unless removed will continue to accumulate

FIG. 6 GRAINER COVERED WITH TENT FOR PROTECTION OF CONCRETE WHILE SETTING

until the grainer is nearly full of salt. It is necessary to "lift" the salt, either by hand when the grainer is nearly filled or by some automatic means as fast as it is formed. The automatic continuous method is the one now generally adopted in grainer plants throughout the United States, not only because it is less expensive than hand lifting, but also because of the difficulty of securing laborers willing to undergo the hard manual labor in the hot steamy atmosphere of a salt plant.

17 Various types of machines, generally designated as "salt rakers," have from time to time been developed for this purpose. They all belong in the class of conveyors, their purpose being to

rake the salt along the grainer bottom and deliver it up an incline located at one end of the grainer. Some peculiar conditions met with in designing machines for this class of work are of interest from a mechanical point of view. Salt rakers, whatever their construction, must be capable of keeping the grainer bottom clear; that is, they must sweep a space of about 150 by 12 ft. continuously, and deliver the salt up an incline, giving the salt an opportunity to drain before being pushed over into the conveyors by which it is transported to the warehouse. The size of the raker practically necessitates the use of iron or steel in its construction, but iron or steel deteriorates

FIG. 7 FORMS AND TENT REMOVED GRAINER FILLED WITH WATER AND FIRST BENT OF BUILDING IN PLACE

rapidly if exposed to the steamy atmosphere of a salt grainer. In fact, it is very common for a half-inch bar of steel to be eaten entirely through in the space of a few months. This corrosion of the steel introduces another element of danger, namely, discolored salt. If a bar of iron or steel is allowed to project down into a salt grainer, a bunch of salt will be formed at the brine level and continue to grow until it is displaced from some cause and that part of the salt in contact with the iron is discolored with iron rust. A very small amount of iron rust will result in streaks of red through the salt piles in the warehouse, depreciating the value of the product.

18 Chains heretofore used in the design of salt rakers have been generally abandoned on account of the liability of breakage. Wheels

have been avoided as far as possible because of the excessive wear on their treads when working in an atmosphere of salt vapor. A chilled cast-iron wheel 8 in. in diameter by 4 in. face, running on a steel track and carrying a load that would not be at all excessive under ordinary conditions, will wear away on the tread during a period of three or four months in the steamy atmosphere of a salt plant to such an extent that there will be practically nothing left except the babbited core and the flanges. Such rapid wear seems to be due, first, to corrosion of the tread, and then to the wearing away of the rust so formed by continued rolling of the wheel on its track. Wheels are therefore to be avoided.

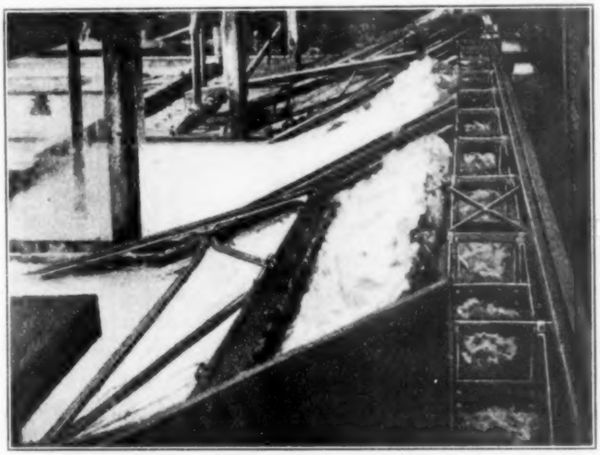

FIG. 8 SWEEPING ACTION OF AUTOMATIC SALT RAKER

19 Another mechanical difficulty in designing a raker for this purpose lies in the fact that the brine surface should be as clear and unobstructed as possible, so that the salt maker can judge from its appearance the condition and grain of the salt that is being produced. Overhead framework is therefore an undesirable feature in raker construction.

20 After much experimenting, passing through many stages of development, the automatic salt raker of today has taken the form shown in Fig. 9 and 10, which are respectively a top plan view and a longitudinal elevation of a grainer equipped with a raker. The raker is driven by a hydraulic cylinder located in front of the grainer.

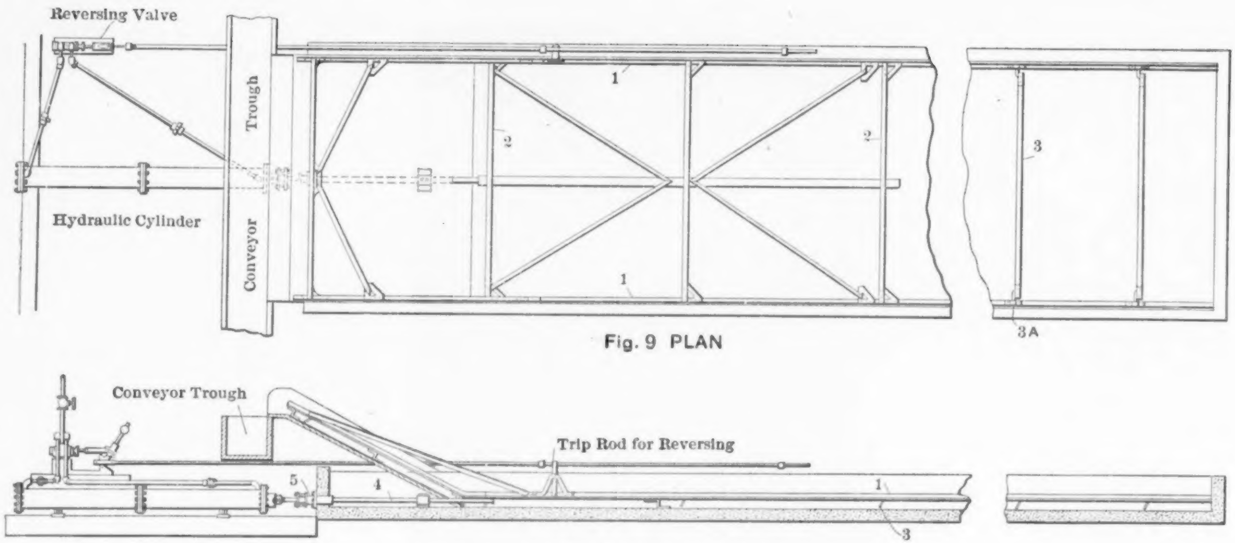

FIG. 9 AND 10 AUTOMATIC SALT RAKER WITH HYDRAULIC DRIVE

21 The sweeps or scrapers of this machine are shown in Fig. 3, underneath the pipes, and the manner in which the salt is pushed up out of the grainer is shown in Fig. 8. The sweeps, by a back-and-forth or reciprocating movement, operate to push the salt toward the end of the grainer on the forward stroke and to feather on the back stroke, forcing the salt with a step-by-step movement along the bottom of the grainer and up the incline.

22 The raker is submerged in the brine and operates underneath the grainer pipes where the air can not get to it except when the grainer is empty. By this simple expedient, corrosion is reduced to the minimum and no difficulty is experienced from red salt. The raker, Fig. 9 and 10, consists essentially of a frame-work comprising two steel angles (1) located within the grainer near its bottom and adjacent to the side walls. At intervals these two angles are connected by cross braces (2) and the frame-work so formed carries a series of feathering scraper blades (3) extending transverse the grainer and supported at their two ends on the two side angles by means of horizontally-projecting rocking pivots or fingers (3A).

23 The scrapers are usually spaced about 8 ft. apart and the entire raker has a back-and-forth movement of about 9 ft., so that each blade travels about 1 ft. ahead of the initial position of the next blade in front. The salt is thus gradually passed step by step to the front of the grainer and up the incline.

24 Hot brine makes an excellent lubricant, and therefore the only support needed for the side angles is a series of flat cast-iron shoes spaced at suitable intervals along the grainer bottom, and it is on the tops of these shoes that the side angles (1) slide back and forth.

25 The rakers may be actuated by a variety of different devices, the hydraulic cylinder, however, affording the best chance for regulation and giving the least trouble in practice. The piston rod (4) passes into the grainer through a stuffing box (5) in which are several rings of metallic packing. The cylinder is usually about 8 in. in diameter by 9 ft. stroke. It makes a stroke in about 2 min., bringing up a load of salt every 4 or 5 min. The cylinder operates at a water pressure of about 80 lb., supplied usually by a steam pump or a motor-driven triplex plunger pump.

26 In the plant illustrated, 10 rakers and 2 reciprocating salt conveyors are operated by one pump. The cylinder water is used repeatedly by returning the discharge from the cylinders to a tank from which the pump takes its suction.

BELT CONVEYORS

27 In conveying the salt from the grainers to the warehouse and discharging it on the warehouse floor many difficulties have been encountered not ordinarily met with in the conveying of materials. On a dry day the salt may be apparently dry and almost similar to granulated sugar in its behavior in the conveyors, but with increased humidity in the weather, the salt becomes soggy and on a very damp or foggy day will even drop brine from the conveyor. If the conveyor is a belt, the salt can be readily cleaned from it on a dry day by means of a diagonal scraper of plate glass, but on a wet day it will stick to the belt so that it is exceedingly difficult to remove it by any cleaning device. Various kinds of scrapers and rotary brushes have been tried but failed.

28 Another difficulty encountered with belt conveyors having iron rollers is excessive corrosion, the under side of the belt becoming covered with iron rust that sooner or later under the sweating of the salt drops rusty water and pieces of discolored salt on the salt piles. It has been found advisable to dispense with iron as much as possible in the construction of belt conveyors for carrying salt, and in the author's opinion, the most satisfactory arrangement is an idler roller made of pepperidge, about 5 in. in diameter and two or three inches longer than the width of the belt. Through the roller is passed a cold rolled shaft, the ends projecting about 6 in. beyond the ends of the roller. The roller is then centered and turned in a lathe.

29 The bearings for the rollers are simply blocks of well seasoned hard maple. Ordinary cup grease is used as lubricant. Bearings of this kind have been run constantly for about three years without serious indications of wear.

30 One difficulty, however, that seems inherent in the use of conveyor belts for handling salt that must be moved in a continuous stream and where the belt is to operate continuously for a long period of time is this: the pepridge rollers reduce in diameter somewhat under the wearing action of the belt and have to be renewed occasionally; but this is not so serious a matter as the discoloration of the salt by particles dropping from the conveyor upon the salt pile. The exact cause of the accumulation of dirty salt along the edges of the belt and the winding of streaks of black salt around the peripheries of the rollers just at the edges of the belt is difficult to account for even after careful investigation, except on the theory

that the moist belt passing through the air in course of a day's run picks up from the air particles of dirt and soot that ultimately work to the edges and wind around the rollers until they have accumulated sufficiently to fly off and drop into the salt pile below.

31 Analysis of such particles showed no traces of rubber and the fact that the belt will run a year or more before wearing sufficiently to necessitate renewal indicates that the dirt does not come from the belt and probably not much of it comes from the wearing of the wooden rollers. It is not practicable to enclose a belt conveyor in an air-tight box; a different style of conveyor has there-

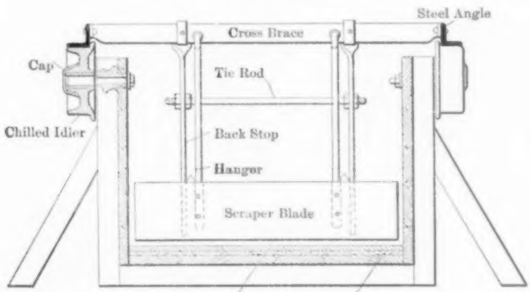

FIG. 11 TRANSVERSE SECTION OF RECIPROCATING CONVEYOR

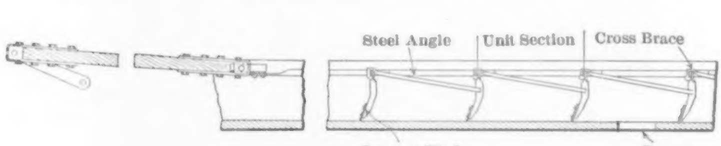

FIG. 12 SIDE VIEW OF RECIPROCATING CONVEYOR

fore been adopted for handling salt under the conditions named, i. e., where a small stream of salt must be taken continuously from the grainer plant and delivered on the warehouse floor. The troubles mentioned together with rapid depreciation, make belts undesirable. These objections apply to continuous delivery of salt in a small stream and not to belts used for cargo loading, or similar uses, where a large quantity must be transported in a short space of time.

RECIPROCATING CONVEYORS

32 A form of conveyor that has been found to be well suited for the continuous handling of salt is shown in Fig. 11, which is a trans-

verse section of a conveyor trough, and in Fig. 12, which is a longitudinal section showing the driving pitman and also the slot through which the salt drops from the conveyor. In this conveyor the scraper blades are usually spaced about 3 ft. apart and the conveyor has a back-and-forth movement of about 5 ft. The principle of operation is the same as that of the rakers previously described.

33 In Fig. 11, the steel side angles of the conveyor are shown supported on idlers with chilled rims. It is not absolutely necessary to use these idlers, as the conveyor works very well if the steel side angles are supported inside the conveyor trough instead of outside

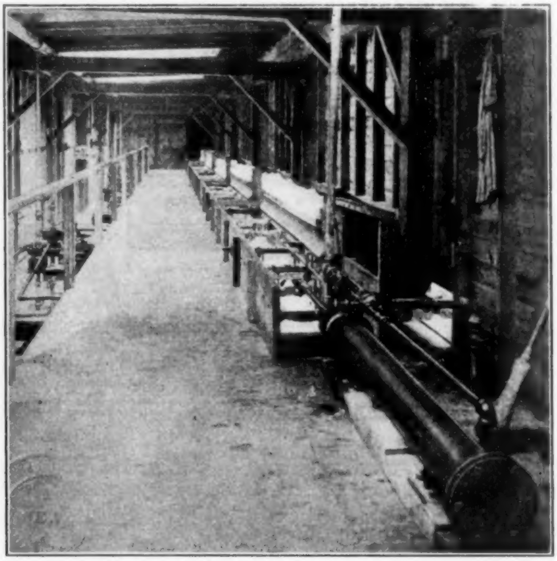

Fig. 13 Conveyor Driven by Hydraulic Cylinder

as shown, and allowed to slide back and forth on angle iron brackets spaced about 6 ft. apart along the trough. The salt dropping into the conveyor from above acts as a lubricant between the two rubbing surfaces. This latter construction has the advantages of cheapness and simplicity, but its disadvantage is the greater liability of particles of dirt getting into the salt. It has been found in practice, however, that the latter disadvantage is more apparent than real.

34 It is preferable to make the scraper blades of well seasoned

hard maple but in some instances they are made of $\frac{1}{8}$ by 6 in. band steel. This type of conveyor must of course be thoroughly galvanized to reduce corrosion as much as possible.

35 The scraper blades and their braces should be arranged in unit sections, as indicated in Fig. 12, so the conveyors can readily be made of any length desired. Such standardization permits reversing the scraping direction of one part of the conveyor if necessary. This type has one important advantage over a continuous belt conveyor

Fig. 14 Apparatus for Loading Under Special Conditions

in that it will scrape in two directions at the same time, if part of the blades are arranged to operate in one direction and part opposite.

36 The bottom of the trough for a conveyor of this type should be as smooth as possible and probably the best lining for the purpose is $\frac{1}{2}$ in. crude plate glass, that is, plate glass which has not been ground or polished. It can be had from any plate glass works at approximately \$0.30 per square foot, and is easily laid by rubbing it into a bed of plaster-of-paris.

37 There is of course a limit to the length that is practicable for a conveyor of this kind, but for salt plant purposes 200 ft. is not

excessive. Such a conveyor, with side angles made of 2 in. by 2 in. by ¼ in. angle, blades of ⅛ in. by 6 in. steel, 2 ft. wide, has been in continuous operation for about four years taking the salt from eleven grainers, each 13 ft. wide by 176 ft. long. The particular conveyor referred to is about 200 ft. long, and the bottom of the trough is lined with hard maple. If it were lined with glass it could probably be successfully operated at a length of about 300 ft. It is driven by a hydraulic cylinder as shown in Fig. 13, or by a crank and pitman, as in Fig. 12. It is probable that this form of conveyor may have other uses than in salt making.

APPARATUS FOR LOADING SALT BARRELS INTO CARS

38 An apparatus found very effective in salt plants for loading barrels into cars, and of interest in connection with locations where

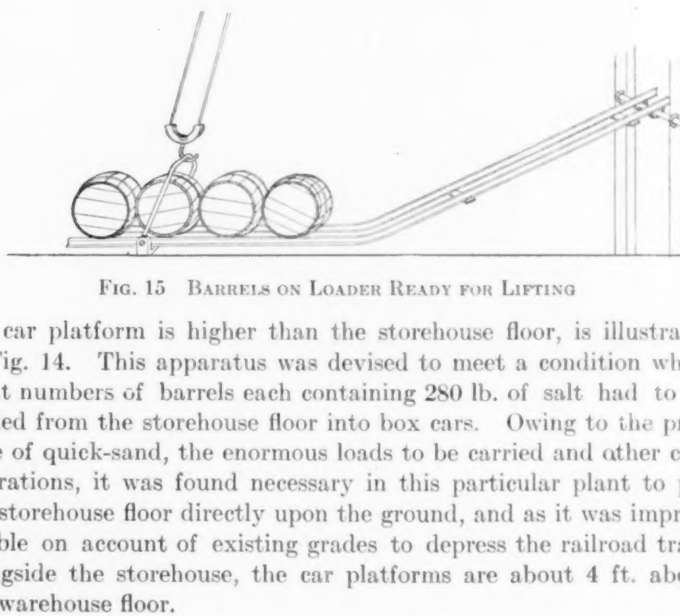

Fig. 15 Barrels on Loader Ready for Lifting

the car platform is higher than the storehouse floor, is illustrated in Fig. 14. This apparatus was devised to meet a condition where great numbers of barrels each containing 280 lb. of salt had to be loaded from the storehouse floor into box cars. Owing to the presence of quick-sand, the enormous loads to be carried and other considerations, it was found necessary in this particular plant to put the storehouse floor directly upon the ground, and as it was impracticable on account of existing grades to depress the railroad track alongside the storehouse, the car platforms are about 4 ft. above the warehouse floor.

39 It is customary in loading a car with salt barrels to put in first a tier of barrels standing on end and then on top of them a tier of barrels lying on their bilges, and this device is well suited to those requirements. An overhead I-beam extending the whole length of the storehouse and paralleling the railroad track (Fig 14)

was provided with a traveling electric hoist. Each loading door of the storehouse was provided with a pair of sockets to receive the rock-shaft of the loading device. Two pieces of light tee-rail are

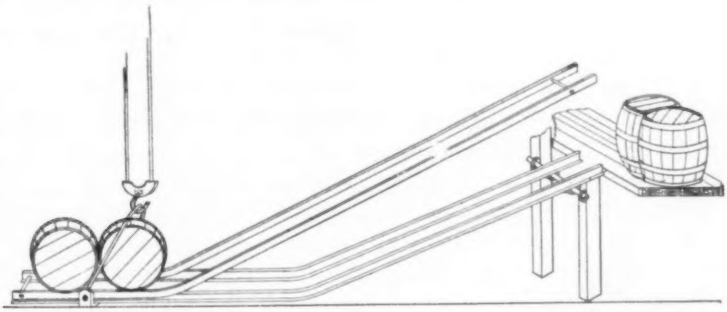

Fig. 16 Loader with Additional Rails for Loading Second Tier

connected at their delivery ends by a cross-shaft and at their receiving ends by a rail to which the electric hoist is hooked. Four barrels are run on the horizontal part of the loader, Fig. 15, and the

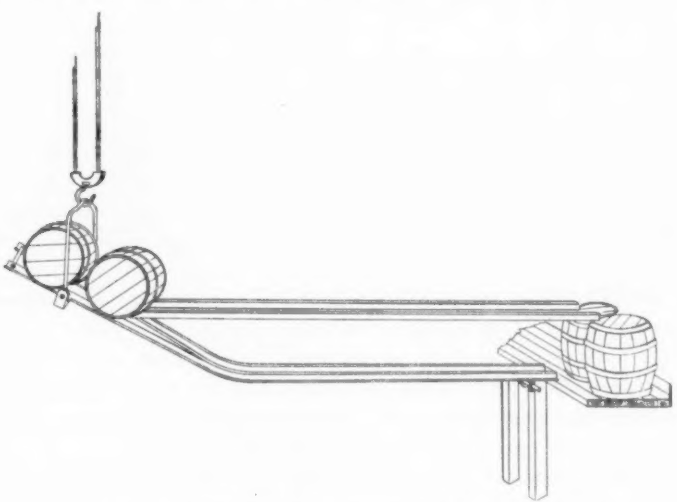

Fig. 17 Delivering Barrels to Second Tier

hoist then raises the end of the loader as in Fig. 14, until the barrels roll into the car. They are then up-ended until the first tier is completed. Then another pair of rails bent at one

end is attached to the first pair so that the free end of the second pair is high enough to deliver barrels on top of the first tier, Fig. 16 and 17. The hoister is simply made to lift higher in delivering the second tier. This hoister is easily moved from door to door of the warehouse, and the electric hoist overhead is easily traversed from place to place as needed.

40 It has been found much cheaper to load barrels into cars by means of this device than to roll the barrels for the first tier from the loading platform of an elevated warehouse into the car and then lift the barrels for the second tier by hand, and this machine makes it practicable to put the warehouse directly on the ground, thereby saving the expense of building an elevated warehouse floor.

41 Neither the reciprocating conveyor previously described nor the loading device are patented, and anyone may make them.

DISCUSSION

Mr. C. F. Hutchings Before lumber became scarce, the "grainer block" was an adjunct to practically every saw-mill in Michigan. By using their exhaust steam in the evaporation of brine, the fuel cost the mills nothing and the cost of salt-production was confined to pumping the brine and handling the salt, which found a ready market at high prices.

2 Salt-manufacturing establishments increased in number, and brine and rock-salt beds were discovered in many states; the market became glutted and prices fell; and cooperage became so scarce that the barrel cost more than the salt it contained. Manufacturers who were not entirely driven out of the market were compelled to adopt more efficient and cheaper methods of producing salt.

3 Finding themselves in this position about six years ago, my company, after investigation, installed mechanical rakers and conveying apparatus, a class of machinery then new to the market. Our apparatus is essentially that described by Mr. Willcox. After correcting the defects incident to a new departure, we found ourselves in possession of excellent salt-handling machinery.

4 The old-style method had required 10 men to lift by hand and store the product of 8 grainers; by the present mechanical methods, 4 men could handle 10 grainers. In our experience a mechanically lifted grainer makes no more salt per day than a hand lift, the raker advantage being altogether one of labor-saving.

5 One of the chief requirements in the design of a salt raker is that

all parts of the machine that come in contact with the hot brine be entirely submerged in it, as the combined action of brine and air will rapidly destroy the apparatus.

6 The pioneer work of Mr. Willcox in exploiting the reënforced concrete grainer, marks an epoch in the engineering department of the salt industry. During the construction of these grainers at Saginaw the salt fraternity looked on with distrust, thinking that the variation in brine temperature would soon destroy the work, but the

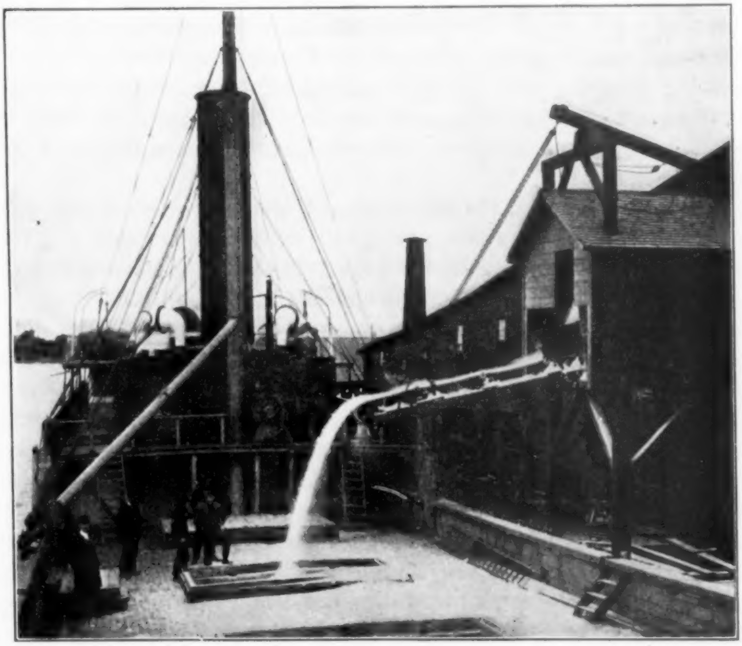

Fig. 1 Loading a Boat with Salt at the North American Chemical Co., Bay City, Mich.

sand joint between grainer and foundation apparently takes care of this point.

7 The scarcity and cost of the high-quality lumber necessary for grainer construction makes the advent of the concrete grainer very welcome. Wooden grainers are hard to maintain and susceptible to many deteriorating influences. For instance, a wooden grainer can be kept water-tight only by continuous use. It will leak if turned out two or three days for repairs or cleaning; this means the loss of

brine and salt for perhaps a week until the leaks are taken up by a fresh deposit of salt.

8 The writer repaired a number of old wooden grainers a few years ago by lining them with 8-oz. duck, saturated with a high-temperature asphaltum compound. The melting-point of this compound is 260 deg., and a grainer seldom shows over 185 deg. The duck, which is $\frac{1}{16}$ in. thick when saturated, was laid with 6-in. laps and all joints covered with the compound and reënforced with strips of saturated duck. After the grainer was well lined another reënforcement of compound was spread over the bottom of the grainer on top of the saturated ducking, and a lining of inch-boards was nailed in the bottom and sides of grainer. This repair job has been running four years and the grainers are still tight, although the original planking is nothing but a bunch of loose fibers which can be pulled to pieces with the bare fingers.

9 As for handling, the belt method is doubtless undesirable with a small but continuous quantity of salt, owing to the liability of the belt-mechanism to collect dirt and spoil the salt. On the other hand, this method is ideal for handling quickly large quantities of salt, a condition exemplified in our salt works at Bay City, where we make salt by vacuum pan as well as by grainer. This salt is carried by belt from the drainage bins which first receive the salt from the vacuum pans to the warehouse bins, a distance of 350 ft., at the rate of 15 tons per hour. The belt is 14-in. operating at a speed of 250 ft. per minute. Five men do the work of handling this 100 tons or 700 bbl. of salt per day; without the belt we employed 9 men.

10 We also have a belt built in a trough across our wareroom floor, with another belt running up an incline at the end to overhang the Saginaw River; under this projecting end a lake-boat of any size can be moored; 30 or 40 men can wheel salt from the warehouse floor in wheelbarrows and dump it into this floor trough, and the salt is forthwith carried by the belt and landed in the hold of the boat. This loading belt is 24 in. wide, runs 300 ft. per minute, and will load at the rate of 150 tons per hour, the rate of loading depending altogether on the number of men at the wheelbarrows. Without this belt we could not load boats with bulk at all, because our warehouse floor is lower than the deck of a vessel.

11 The net result of our installation of salt handling machinery is as follows:

On grainer raking outfit, saved the labor of 6 men.

On vacuum pan outfit, saved the labor of 5 men.

A total saving of the labor of 11 men on handling about 1000 bbl. of salt per day, a cash saving of nearly two cents per bbl. of salt.

The interest charge on these improvements and the repair account come to a very small fraction of a cent per barrel.

THE AUTHOR As showing the relative advancement in the development of special salt works machinery in America and abroad, it is of interest to note that the German Government recently sent a commission of salt experts to this country, and as a result that Government has adopted this system and has authorized the building of a large salt works at Schöningen, which is in its essential features of design and equipment practically a duplicate of the plant described in this paper.

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No. 1223

INDUSTRIAL PHOTOGRAPHY

By S. Ashton Hand, Cleveland, O.

Member of the Society

Photographs of machinery, interiors of shops, products of machines, processes of manufacture, etc., are generally made to aid the selling department of an establishment in disposing of its product.

2 Sometimes the photographs themselves are used as an advertising medium, but in the majority of cases half-tones are made from them for use in catalogues, or for illustrations in trade journals. Catalogues have of late years developed into veritable works of art, and their preparation calls for photographic work of the highest order.

3 To this end it should be the aim of the photographer to produce prints that will require the least retouching when used for making half-tones, and this for two reasons: First, the retouching of prints for half-tone work is quite expensive; and second, the print that requires the least retouching gives much the best results in the finished half-tone. A print which requires very little retouching to produce a first class half-tone is a good one for all other purposes, but a print good enough for all other purposes may be a very poor one from which to produce a first class half-tone.[1]

4 Nearly all industrial establishments are equipped with a photographic outfit of some kind, and in some instances an experienced photographer is in charge; but in the majority of cases one of the draftsmen must take care of all the photographic work of the establishment, and it is in the hope of aiding some of the latter that this paper has been prepared.

[1] The reproductions in this paper were made from photographs upon which there was no retouching whatever.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

APPARATUS

5 The camera should be a strong and serviceable one having a long bellows with very little cone. In fact, one with a perfectly straight bellows is best, as it allows greater adjustment of the lens board without danger of the bellows folds cutting off any of the object. The vertical and side swings should be ample.

6 The camera need not be larger than 6½ by 8½ in., and should not be larger than 8 by 10 in., as anything over this size is cumbersome to handle, and requires a very expensive lens and a great deal of skill to operate.

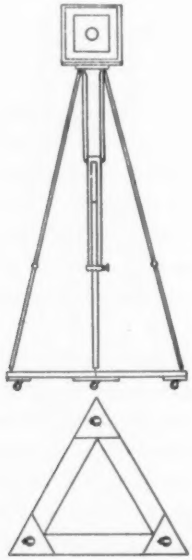

Fig. 1 Tripod with Triangular Base

7 If large prints are wanted, bromide enlargements can be made up to any reasonable size, and if for any reason large direct or contact prints are wanted, a slightly enlarged positive can be made from the negative, and a negative as large as wanted can be made from the positive. This procedure has its advantages, as it is often possible to correct in a great measure any errors in exposure or development, and many errors in lighting and position.

8 The writer never uses a camera larger than 6½ by 8½ in., and

has produced many excellent enlarged negatives up to 24 by 36 in., by the method above mentioned.

9 The tripod should be solid and stiff with the fewest possible joints. An excellent base for use with it is a triangle with sides about 36 in. long, and with a roller or caster under each point as shown in Fig. 1. With the tripod mounted on this arrangement, the camera can be moved any distance or in any direction without material change in level.

10 The lens should be the best obtainable, and too great emphasis cannot be placed on its being of long focus. Never under any circumstances should its focus be shorter than the diagonal of the largest plate with which it is to be used. It should be capable of rendering sharp definition from corner to corner of the plate when using a comparatively large diaphragm. A lens of this character will render the focusing much easier, and will enable the exposure to be made in the shortest possible time.

11 The plates should not be the most rapid made, as the emulsion with which these are coated is not generally rich enough in silver to give printing density for anything but portrait work, and also because the timing of the exposure must be very exact. Unless both exposure and development are just right, the negative will not be "snappy" enough to produce a good bright print.

12 Very slow plates take long exposures, and unless skillfully handled in development will produce prints with entirely too much contrast. Plates of medium speed are the best and should be of the kind known as "double coated" or "non-halation." Plates of this kind are first coated with a slow emulsion, and after drying are again coated with a somewhat faster emulsion. Plates so coated allow of very great latitude in time of exposure.

13 If interior views are to be made where windows and other openings to the light have to be faced, then the plates should be coated on the back with a compound known as "backing." This will prevent to a great extent halation or blurring of the high lights caused by reflection of light from the surfaces of the plate under the emulsion. This "backing" should be washed off with a damp sponge before development.

PREPARATION OF THE SUBJECT

14 If a machine is to be photographed, it should be painted with a finishing coat of drab paint, which may be designated as "mouse

color," and the paint should be so mixed as to dry absolutely "flat," that is, without any gloss whatever. If parts underneath the machine or in shadow are wanted to be shown, they should be painted a lighter shade than the more prominent parts, and the deeper they are in shadow the lighter they should be painted, even in extreme cases blending the color gradually into a white. All brightly polished parts should be daubed or rubbed over with a handful of soft putty to dull the brightness.

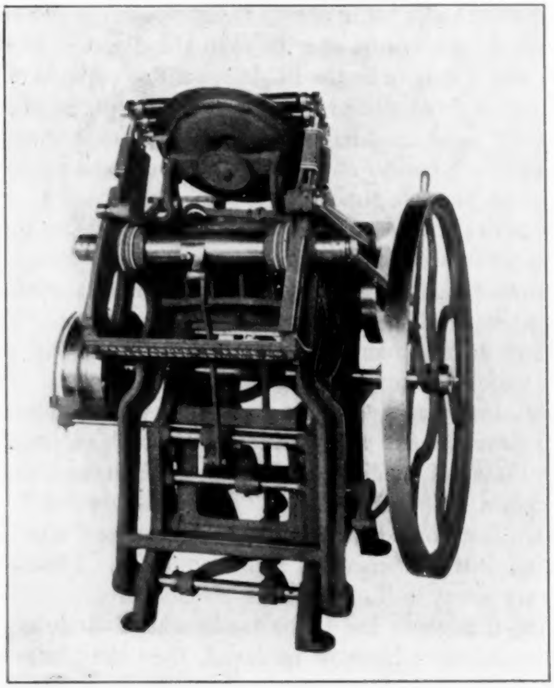

Fig. 2 Details Brought out and Shadows Avoided by Preparation of Machine Parts Before Photographing

15 Unless these precautions are taken, the parts in shadow will show very dark in the photograph, and if very close together will be seen only as one shapeless mass, and the bright spots will show chalky white with very black lines and little or no detail. If letters or figures cast on any part of the machine are wanted to be shown, daub them with white paint from the end of a finger. Rubbing with chalk will give them a very rough appearance.

16 It must be borne in mind that all high lights and shadows are greatly intensified in photography, and that a sensitive plate that will register all the gradations of tone seen by the human eye has yet to be made.

17 Fig. 2 and 3 are illustrations of machines that were properly prepared for being photographed.

18 If possible, it is best to photograph a machine before it has been run, otherwise oil from the bearings will seep out on the paint

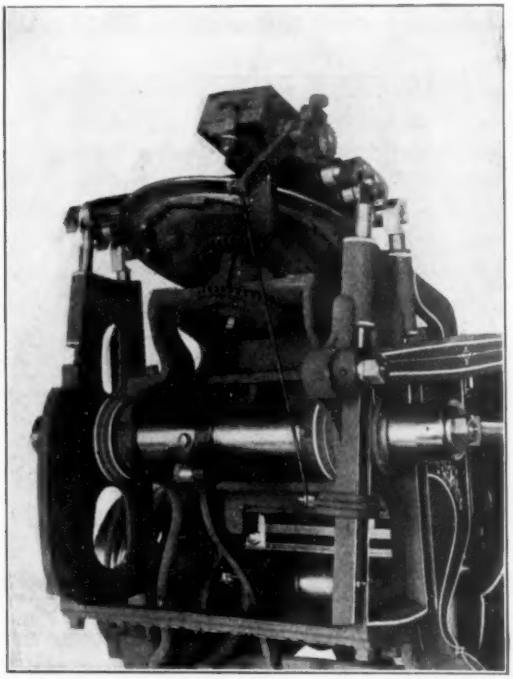

FIG. 3 DETAIL OF FIG. 2

and leave dark and glossy spots which will look badly in the photograph. If the machine is to be run before being photographed, then it should not have its finishing coat until after the run or test is over. Before the finishing coat is put on, all the bearings should be thoroughly flushed with gasolene and the whole exterior cleaned with it to remove all oil.

LIGHTING AND POSITION

19 Machinery should never be photographed out of doors or under a skylight, as there is too strong a top light which causes deep shadows.

20 The light should preferably come from the north, and should fall on the machine at a downward angle of about 20 deg. from the horizontal. Cross lights from other windows should be avoided by pulling down the shades or tacking up heavy paper. Cross lights make a confusion of shadows and obliterate certain lines, giving the machine anything but a natural appearance. If necessary to photograph the machine by other than northern lighting, then make the

FIG. 4 DISTORTED VIEW MADE BY POINTING THE CAMERA UPWARD

exposure when the sun is overhead. If the exposure must be made when the sun is shining through the windows at any considerable slant, tack cheese cloth over the windows to diffuse the light.

21 A machine should never be photographed directly from the front, which will make it appear too flat. For depth, the camera should be placed enough out of center to show a little of one side of the machine and high enough to show a little of the top.

22 A background of heavy drilling, either white or very light in color, should be hung not less than 6 ft. back of the machine. It should be of ample size—large enough so that the camera can be

moved where wanted and still show the background behind every part of the machine. If there are folds or wrinkles in the background, have a man at each side take hold of the edges and shake the curtain slowly and gently during the whole time of the exposure. This will prevent the folds or wrinkles from showing in the photograph.

23 Shop floors are dark in color, and if a machine is photographed directly on the floor it is often puzzling to know where the lower part of the machine ends and the floor begins. Therefore a floor cloth of the same color and width as the background should be used. It should be deep enough to extend from 4 to 6 ft. in front of the

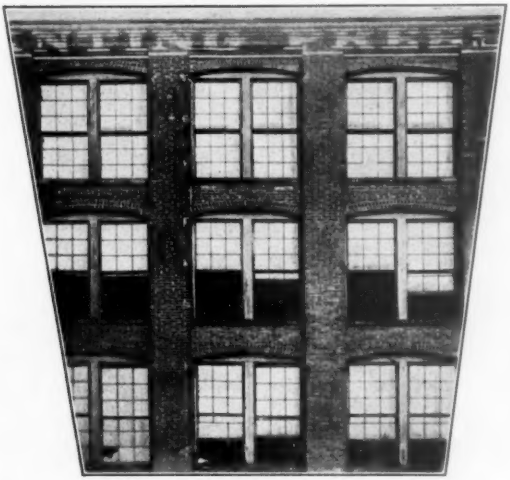

Fig. 5 Distortion of Fig. 4 Corrected in Reproduction

machine and under it and to the background. This will define the lower parts of the machine, and also reflect the light upwards, softening the shadows. Instead of a floor cloth, sheets or strips of light colored paper can be used, but be sure there is no pronounced red or yellow, as such colors are non-actinic and will show black in the photograph.

FOCUSING

24 Never focus on the center of the ground glass, as this will give the point of sharpest focus of the lens and what is wanted is the average focus; therefore focus at a position midway between the

center and the edges of the ground glass. Get the nearest parts of the machine in focus. Small diaphragms will sharpen up the distant parts.

25 Sometimes a better effect can be obtained by pointing the camera slightly downward, but if at any time the camera is used in any other than a level position, the ground glass should be brought to a vertical position, otherwise the result will be distorted lines.

26 Fig. 4 shows a distorted view of a part of the side of a building, made by pointing the camera upward. If the photograph of a

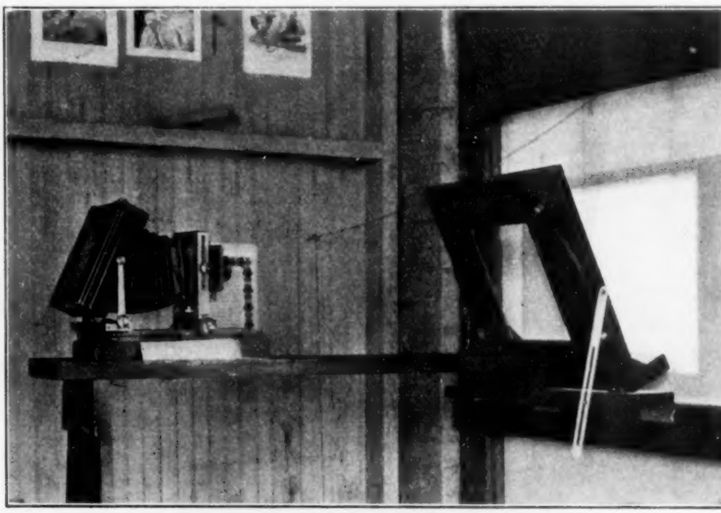

Fig. 6 Camera and Negative Holder Arranged so as to Correct Distortion Shown in Fig. 4

machine shows such distortion, and for any reason it can not be photographed again, a negative can be reproduced eliminating the distortion, by placing the negative in a frame tilted at such an angle that the narrowest lines are nearest the lens, and making a positive in the camera, tilting the ground glass at an equal angle, but in the opposite direction. A negative can be made from this positive, as shown in Fig. 5, which was actually made from a negative reproduced in the above manner from that used for Fig. 4. Notice how much the top of the negative had to be enlarged to bring the lines parallel.

27 Fig. 6 shows a camera and negative holder in the proper position for this operation.

28 If the machine to be photographed is a long one, requiring a raking view, use the horizontal swing to bring that part of the ground glass on which the image of the farthest part of the machine appears, farthest away from the lens. This will even up the focus and make it possible to use a larger diaphragm, shortening the time of

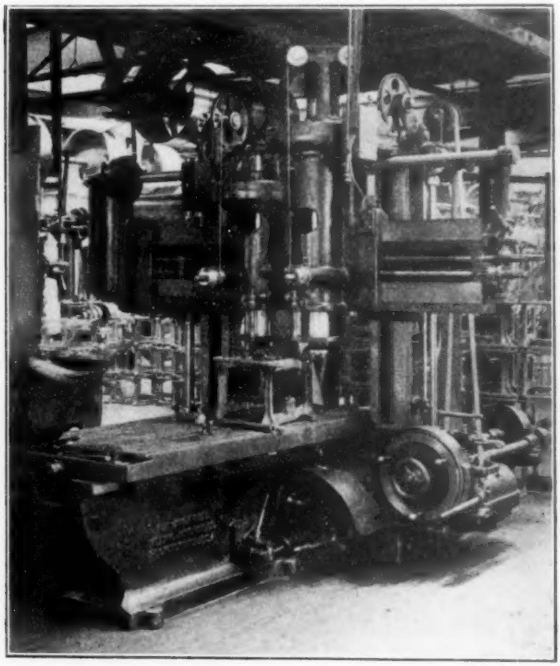

Fig. 7 Details of a Dark Shop Brought out by Long Exposure

exposure, and also extend the vanishing point to a greater distance, giving it a more normal perspective.

29 If there are perceptible vibrations to the floor on which the photographing is done, get three pieces of harness felt $\frac{1}{2}$ in. thick, and two or three inches square. Place one of these on the floor under each leg of the tripod, and they will absorb all ordinary vibrations and keep the camera steady.

EXPOSURE

30 Exposures should always be ample, as an under-exposed plate can never be made to show that which the light has not impressed upon it (although it can be greatly helped by skillful development), but a moderately over-exposed plate can easily be treated in development, or even afterwards, so as to yield a first class print. Fig. 7 is an illustration of details brought out by long exposure in the dark part of a shop.

Fig. 8 Photograph from Negative Taken without Special Precautions against Strong Light—Strongest Light at Far End

31 If in doubt as to the correct time of exposure, make a guess as near as possible. Suppose your guess to be four minutes, then put a loaded plate holder in the camera and draw the slide so as to expose two inches of the plate and make an exposure of two minutes; cap the lens, draw the slide out two inches more, and make another exposure of two minutes. Repeat this, drawing the slide two inches at a time, until the whole plate has been exposed.

32 If the plate is an 8 by 10, there will be five parts having respective exposures of 10, 8, 6, 4, and 2 min. each. Develop this plate, and it will be easy to tell which part has had the proper exposure, and from the position of this part the time can readily be found.

INTERIORS

33 In photographing interiors, avoid pointing the camera toward windows if possible, but if this cannot be avoided, then cover the windows with heavy drilling or thick wrapping paper, fastening it

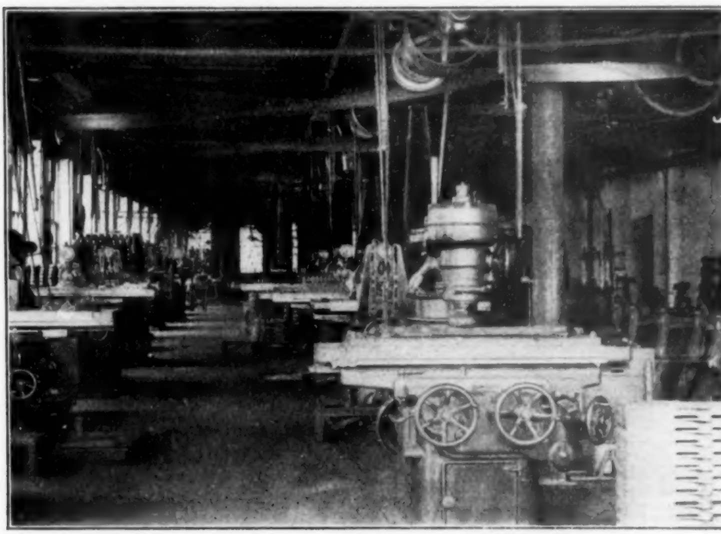

FIG. 9 SAME CONDITIONS AS IN FIG. 8, BUT WITH STRONG LIGHT IN FOREGROUND

well around the edges, so that no bright margins of light are visible. After the exposure has been made, the window coverings can be removed and an additional exposure of a fraction of a second can be given. This will give the windows a natural appearance and will often show objects on the outside. Interiors can be photographed without these precautions, but skillful work will be required to make good negatives.

34 Fig. 8 and 9 were made on a very bright day when snow was on the ground, and the light coming in the windows was intensely white. As the negatives were wanted in a hurry, no precautions

were taken to soften or stop out the light at the windows. The far end of Fig. 8 was a southern exposure, and the sunlight was streaming in at the windows. Fig. 9 is a view in the same room taken from the south end, where the light was so intense that the milling machine in the foreground appears light in color, although it was painted a dark steel color.

35 Fig. 10 shows work in process in a special machine. This was made on the same day as Fig. 8 and 9, and with the camera pointed directly at the light. To work successfully under such conditions, the photographer must know to a nicety just how long a

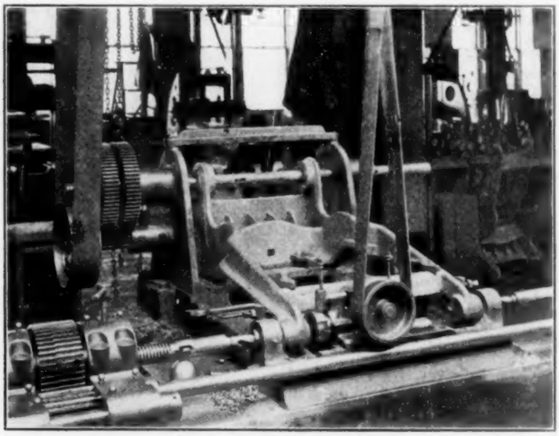

Fig. 10 Photograph on Same Day as in Fig. 8 and 9. Camera Pointed Directly towards the Light

certain kind of plate may be exposed before halation takes place, and just how to get the best results in development from the shortest permissible exposure.

COPYING

36 In copying drawings or other subjects in black and white, it is necessary to use a very slow plate, give the shortest possible exposure, and use a concentrated and well restrained developer. Unless this is done, the lines of the drawing will not be clear and sharp.

37 If a copy is to be made from a blue print, it will be necessary to bleach the print in a weak solution of ammonia and water, and

after a thorough washing, to immerse it in a weak solution of tannic acid. The part that was formerly blue will now be a rich purplish brown. The necessity for this treatment is that blue is an actinic color, and a negative made from such a print will have very little contrast; while brown is a non-actinic color, and a negative made from a print of that color will have plenty of contrast.

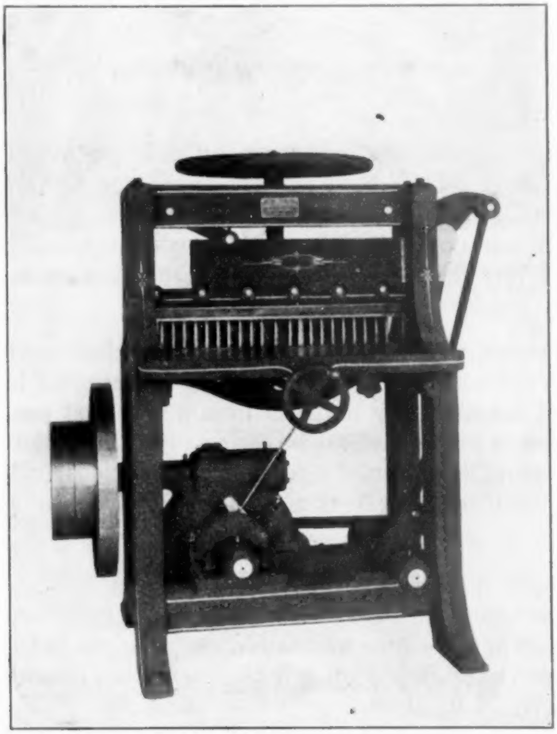

Fig. 11 Photograph of Paper Cutting Machine, Showing Clutch Mechanism to be Brought out by Photographic Methods

ENLARGING NEGATIVES

38 A negative can be reproduced in a larger size by first making a positive in the copying camera, and then making a large negative from the positive by the use of the same instrument.

39 If a negative is enlarged to many times its original size, a granular effect will be noticed. This is caused by magnification of

the emulsion structure which is made up of countless thousands of hills and valleys. This granular effect can be eliminated by slightly over-exposing and greatly over-developing the original negative, and then reducing it to the proper density. The positive should have the same treatment.

40 Reduction does three things:

a It reduces or clears the shadows faster than the high lights. Therefore over-exposure is resorted to in order to increase the density of the shadows in proportion to the high lights, so that they shall bear proper relation to each other after reduction.

b It thins the density of the negative or positive. Therefore over-developing is resorted to in order to have resulting density after reduction.

c It cuts down the hills to the level of the valleys, so that very little if any granular effect is noticeable when the emulsion is magnified.

41 In reproducing negatives either in the original or a larger size, there is a splendid chance for what may be termed "jockeying." A brilliant negative may be made from a very flat one, and vice versa; errors in perspective can be corrected by the method shown in Fig. 5; unequal lighting can be corrected by judicious shading during exposure, and various effects secured by an ingenious operator.

X-RAY OR GHOST PHOTOGRAPHS

42 When an illustration is wanted to show clearly some hidden interior part of a machine in relation to and more distinctly than other parts, the usual procedure is to have a wash drawing made in India ink, from which the half-tone is produced. This method is always expensive, and the results are often very unsatisfactory.

43 Fig. 11 shows a power-driven machine for cutting paper, in which the power is transmitted through worm gearing and a positive clutch, all of which is enclosed in an oil tight case, as shown at the lower left hand part of the machine. An illustration showing the worm and worm wheel in mesh was wanted.

44 An engraver was called in, who said he could do the job in a satisfactory manner. Blue prints showing both details and construction were furnished him as a guide to size and shape of the

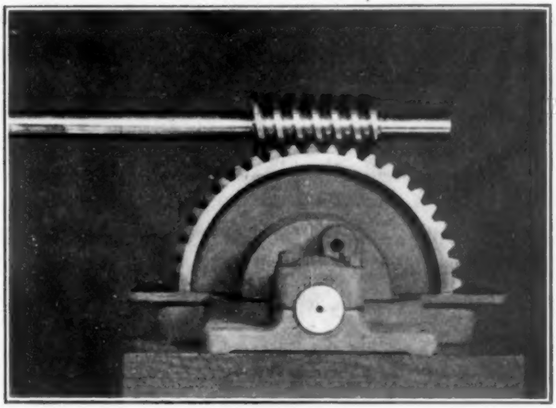

Fig. 12 Photograph of Worm and Gear with Upper Part of Case Removed

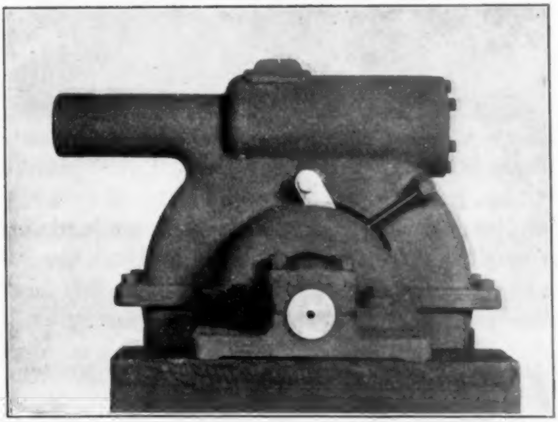

Fig. 13 Exterior of Case to be Photographed on Same Plate with Fig. 12

various parts, and their appearance when assembled. After repeated attempts, his results were not satisfactory, and it was decided to rely entirely upon photographic methods for the illustration.

45 The case and its contents were removed from the machine, and mounted on a box. The upper part of the case was taken off, leaving the worm wheel and the clutch collar exposed to view. The worm and shaft were removed from the upper part of the case and placed in their proper position in relation to the worm wheel, as shown in Fig. 12. A dark back ground was placed in the rear and an exposure was made. After the exposure the cap was put on the lens, the worm and shaft taken away, and the upper part of the case put in position as in Fig. 13. A light back ground was sub-

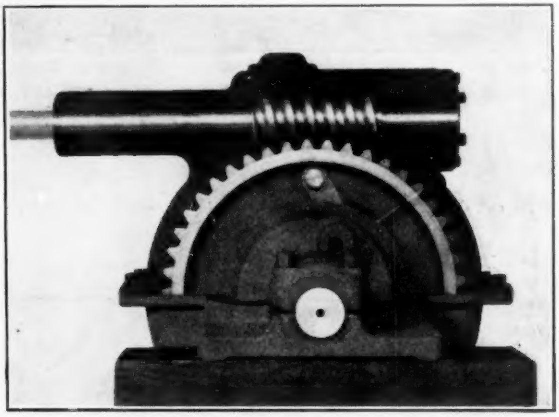

FIG. 14 RESULT OF EXPOSURES FIG. 12 AND 13 ON SAME PLATE

stituted for the dark one, and another exposure made on the same plate, the result of which is shown in Fig. 14.

DEVELOPMENT

46 Of the art of development much has been written, and more has been said, but the fact remains that the only adequate teacher is experience. After many years of experience, the writer would hesitate to attempt to tell in writing how to handle a sensitive plate in development. A few hints however, concerning the behavior of certain mixtures of developer, may be of service.

Too weak a developer makes a flat thin negative.

Too concentrated a developer makes a negative with too much contrast.

Under-developing makes a negative lacking in printing density.

Over-developing makes a thick dense negative requiring a very long time to print.

An under-exposure should be developed with a diluted developer.

An over-exposure should be developed with a concentrated developer well restrained with bromide of potash.

Developers, like artists' colors, should be mixed with brains.

Choose a moderately slow developer, learn how to use it, and don't let any one persuade you to change it.

PRINTING

47 Printing can be done in so many ways and on such a variety of papers, and the solutions necessary for developing and toning require such care in compounding and handling, that it can in most instances be better and more cheaply done by a professional.

DISCUSSION

Dr. C. J. H. Woodbury I am a photographer by proxy only, but will give my experiences with bright metal work, in having photographs made of machinery.

2 Many years ago, the late John C. Hoadley, a member of the Society, extensively advertised the portable steam engines made by him at Lawrence, Mass., by means of photographs for which the finished metal work was whitewashed, giving excellent results.

3 In photographing silver, shaking a bag of cheese cloth containing face powder near the articles until the white dust has settled upon the object and cut off the glare, has given more satisfactory detail than the putty process, as putty contains yellow tints not visible to eye, but giving in a photograph noticeably darker effects than those of silver. Where silverware is extensively photographed, the articles are arranged, and the camera focused; then they are removed to a refrigerator, and chilled; on being replaced in position the moisture from the air condenses upon the cold surfaces, and the photograph is taken before it accumulates into drops.

4 Photographers claim to obtain the best results with small diaphragms, moderate lights and long exposures. Interiors containing much detail can be photographed by darkening the room, drawing the curtains, closing the blinds, and giving an exposure of from one to two hours, with a small diaphragm, after which the camera is closed, and the blinds are opened and the curtains raised to admit the light, and a snapshot is taken, using a larger diaphragm; the result will be a photograph revealing detail to an extent not possible with single exposure.

5 A public building decorated for a civic celebration with flags, bunting and many incandescent lights was successfully photographed by first taking an insufficient exposure during the day, and then without moving the camera taking another exposure during the electric illumination in the evening, with the result that the building was faintly shown in the photograph much as it appeared to the observer in the evening.

Mr. Henry B. Binsse I would like to take issue with the author about one point—he says that the machinery should be painted a drab color. Of course I have not had as much experience as he, but the best results which I have ever obtained have been from painting the machines a salmon pink. The effect of drab color was to make a machine look flat, while if painted a warm color the parts stand out more and a clearer picture is secured.

Mr. Charles W. Hunt In connection with Mr. Hand's paper it may be of interest to submit a record of some experiments made by the writer for the purpose of estimating the proper time of exposure for dry plate photography. The following tables are based on this series of experiments. As a preliminary step the altitude of the sun was calculated, for each hour of the day from sunrise to sunset, on the first day of each month in the year, and the results plotted.

2 In June 1905 a series of exposures was made with a Watkins exposure meter at each hour of the day, and a tentative table of the relative exposure time for each hour from sunrise to sunset was made. Using this tentative table a series of similar exposures was made with dry plates. The plates were each developed the same length of time and in the same strength of developer. From these tests the tabular time was corrected.

3 A table was then made giving the estimated time of exposure for each hour of the first day in each month in the year, basing the time of exposure largely upon the tests and the altitude of the sun in the different months. During the ensuing year this table was tested from month to month, and revised as experience indicated, in order to get the best attainable negative at any hour of the day in any month of the year. The present table is derived from the results of the above tests, with the formulae corrected to correspond to exposures made on Eastman films of the current year 1908.

4 The time for a theoretically perfect exposure, that will result in the best printing negative that the subject will give, cannot be expected from any formula that takes into consideration only the most prominent factors affecting the problem. These rules may, however, be expected to give a reasonably close approximation to a perfect exposure.

5 In making exposures, where it is unusually important to secure a good negative, and the exposure cannot be repeated, proceed as follows: Compute the proper exposure by the formula. Then give three exposures: (1) The first exposure with time as computed; (2) The second with one-half the computed time; (3) The third with double the computed time.

6 For less important cases, but where great uncertainty exists as to the proper time of exposure, proceed as above, but make only two exposures, the slowest and the fastest, omitting the computed time exposure. The latitude of the plate will give a satisfactory negative if the theoretically perfect time of exposure lies within very wide limits.

7 An exposure should not be made in a fog, and in hazy weather only of nearby subjects. Good negatives may be made during a shower if the weather is otherwise clear. Generally, if contrast in the negative is desired, "underexpose;" if definition in the shadows is wanted, "overexpose." When in doubt, it is safer to "overexpose." Stops number 64 or 128 are excellent for general outdoor exposure; number 32 or 16 for indoor work. If it is desirable to emphasize a specific part of a machine focus carefully with a large stop and shorten the exposure to correspond with the stop.

8 The following formulae and tables are based on normal light conditions and ordinary subjects. If either or both are abnormal, the operator must make an allowance in the duration of exposure as computed by coefficients from Tables 1, 2 and 3.

TABLE 1 COEFFICIENTS (A) FOR PHOTOGRAPHIC EXPOSURES IN THE LATITUDE OF NEW YORK

Month	Hour of the Day				
	7 to 8 or 5 to 6	8 to 9 or 4 to 5	9 to 10 or 3 to 4	10 to 11 or 2 to 3	11 a.m. 12 m. 1 p.m.
January—December	..	2	4	6	7
February—November	..	4	5	7	8
March—October	2	5	6	8	12
April—September	4	6	9	12	16
May—August	5	8	12	18	28
June—July	7	10	16	24	32

TABLE 2 WEATHER COEFFICIENTS (B)

Clear, sunshiny weather	1.
Floating, white fleecy clouds	1.
Overcast, but a light day	1.5
Cloudy, dull day	2.
Lowery, heavy clouds	4.

TABLE 3 SUBJECTS: COEFFICIENT (T)

Shop interior, dark and poorly lighted	1000.
Shop machinery, fairly well lighted	400.
Shop machinery, placed near a good window light	150.
Machinery under sheds with one side open, or covered areas	15.
Machinery outdoors to give details in the shadows	2.
Machinery outdoors, general views	1.5
Groups, or portraits outdoors	1.
Buildings and nearby landscape	1.
Distant structures or landscape views	0.5

TIME EXPOSURE

9 Assume a stop suitable for the subject and call it H; then the seconds to expose will be:

$$\frac{H \times B \times T}{32 \times A} = \text{seconds exposure for an } H \text{ stop} \qquad (1)$$

BULB EXPOSURE

10 A "quick" bulb exposure is a time exposure of about $\frac{1}{5}$ to $\frac{1}{4}$ sec. To compute the number of the lens stop use the formula:

$$\frac{8 \times A}{B \times T} = \text{stop for a "quick" bulb exposure} \qquad (2)$$

Mr. H. H. Suplee I have used with a good deal of success flashlight powder in connection with daylight, to emphasize the dark part of a machine, where it is impossible to get proper light on small

parts; a small flashlight discharged out of the field so as to reflect the light on the machine will supplement time-exposure and bring out the parts which would not otherwise be visible.

2 My own experience is that it is almost impossible to compute the time of exposure with any accuracy when there are so many variables. The amount of light has to be estimated always, and in my opinion the best plan is to use a moderately slow plate and give a long exposure. If the plate is developed with a long development, you will get detail which you could hardly calculate or compute by attempting to work at any definite time of exposure.

3 Photography has been put to very important uses in engineering work lately, in connection with moving picture machines. We saw last night moving pictures showing the operation of a modern flying machine. They have also been used to show the operations in the workshop, by mounting the machine on a car, and running the car at moderate speed, and the picture machine at full speed, through the shop, the result being a lesson on shop practice, which can be used in a course of lectures more effectively than any other possible means.

4 Several years ago I suggested the possibilities of the moving-picture machine as a means of engineering investigation. It is possible to run the machine very rapidly and get a succession of pictures that represent a very short duration in making the strip. This film can then be run through the picture machine at a speed much slower, with the result that an operation so quick as to be imperceptible to the eye is slowed down so as to be readily examined. I have made no experiments, but I think it is possible, for instance, to photograph a fracture of a piece in the testing machine, occupying only a few seconds at the critical moment, so rapidly in an artificial light on a continuous film, that it could later be thrown on the screen and slowed down so that the whole sequence of rupture would be plainly visible. Many other applications will suggest themselves for studying phenomena altogether too rapid to be examined by the eye. I believe these matters are likely to be applied to the laboratory, and possibly to workshop operation.

5 Another use is that of keeping a record of the progress of work, a record which cannot be disputed; this is often applied by contractors and builders day by day, to keep a record of the operation of buildings, etc.

6 Another use of photography, which I employed a number of years ago with a good deal of success, is for preserving a record of valuable drawings in a limited space. A full set of complete double

elephant drawings of a whole line of machines were photographed down to $6\frac{1}{2}$ by $8\frac{1}{2}$ in., and a set of prints were made, which formed a parcel that could readily be put away in a safe deposit drawer. In the case of the destruction of the originals by fire, these could be enlarged back to the original dimensions, and thus a perfect fire insurance is obtained for a valuable set of drawings.

Dr. C. J. H. Woodbury. As an example of the use of photography to represent the operation of machinery, the makers of a magazine loom which contained many new and important features had a series of moving pictures taken showing one of these looms in operation, and half-tone plates were engraved from these continuous negatives.

2 These half-tones were printed on a series of small cards hinged together and placed in a small portable device resembling a stereoscope so that by turning a crank the pictures were revealed in rapid sequence, and one could see the loom operating at normal speed, or by turning the crank very slowly, the various operations of the mechanism could be noted in a way not possible of observation when the machine was in its normally rapid operation. Many of these sets of illustrating devices, which were merely those used for a long time in moving-picture shows, were placed in the hands of salesmen and were doubtless the cause of the rapid introduction of this loom.

3 At the other extreme, photography has been used to analyze very slow motions, as for example, a growing plant, which was photographed day after day, the results when put in a moving-picture machine giving the method of extension of the plant. About twenty years ago, the late Col. W. E. Barrows, a member of the Society, built a mill at Willimantic which contained a great many novel and original features, and weekly photographs were taken and sent to members of the board of directors to illustrate the progress of the work.

4 When one of the recent half-encyclopedias and half-dictionaries was in process of preparation, a great many of the more valuable manuscripts were photographed down to about the size of a postage stamp for a page, and the negatives of the photographs were kept in a safe-deposit vault. These instances merely indicate that the dry-plate method of photography has placed in the hands of laymen without the expensive plant and skill of the old-time photographer, the ability to show things, as the author stated, as "they are" and

to give a true record of the progress of work or the existence of any forms of machines or other objects.

Mr. Ambrose Swasey complimented Mr. Hand on his photographs of machinery and interiors and pronounced the paper of very great value to the Society.

The Author The use of face powder as well as of refrigeration, in photographing articles of silver, is new to me, but I think either would be excellent for the purpose.

2 Whether small diaphragms, moderate lights and long exposures produce the best results, will depend on the amount of contrast in the subject to be photographed. Size of diaphragm and length of exposure must be governed entirely by conditions and no set rule can be given governing their relation to each other.

3 The methods of making photographs of illuminated buildings and of interiors, have been successfully practiced for years.

4 Painting machinery a salmon pink would be all right for photographing in a very subdued light. I have never had any trouble in getting plenty of contrast with machinery painted a drab color. In fact, I have to give long exposures to avoid too much contrast.

5 Mr. Hunt has gone into the matter of exposures very thoroughly, but he has left out of his table of weather-coefficients one very important condition of light, and that is its color. A day may be clear and sunshiny, but if the light is yellow an exposure of four to ten times that deduced from his formula may be necessary. Exposure-tables are all right for the first guess, but any one of the many factors to be taken into consideration may so upset all other calculations that experienced photographers have never used them to any extent.

6 The use of flash-powder in connection with daylight exposures to emphasize dark parts of machines, is all right, providing there are no projecting parts to cast shadows. Shadows cast by flash-light are very intense.

7 For the successful use of a moving-picture machine in the shop, the shop must be a very well-lighted one, or a lens with an enormously large opening in proportion to its focal length must be used.

8 I have been asked if I have had any experience in showing by photography whether buildings were plumb or not. An interesting experience that I had several years ago will serve as an answer.

9 While the Bourse Building in Philadelphia was in course of construction, a man asked me to photograph the steel structure which had just been erected. I found he was one of the kind of men who write for the daily press on all sorts of civic questions, signing themselves "Pro Bono Publico," or something of like nature. By great difficulty we obtained a position on top of a building where we could get a good view of the steel work. After getting my position and focusing the camera, I asked what he wanted the photograph to show. He said the steel work was out of plumb and he wanted the photograph to show it. I turned to him and said, "Which way do you want it out of plumb? I will make it either way you want it."

No. 1224

THE PHYSICAL PROPERTIES OF CARBONIC ACID AND THE CONDITIONS OF ITS ECONOMIC STORAGE FOR TRANSPORTATION

By Prof. Reid T. Stewart, Pittsburg, Pa.

Member of the Society

The accompanying tables and charts show in condensed form the results of a recent investigation into the Physical Properties of Carbonic Acid. They furnish the data necessary in investigating the strength and safety of existing carbonic acid cylinders, and in the design of new cylinders on a safe and economic basis. The value of these tables will be apparent when it is considered that each of the hundreds of thousands of the cylinders now in use becomes, when charged, a reservoir of stored energy, which would in all probability cause loss of both life and property should rupture occur.

2 These tables also furnish sufficient data for determining the conditions that render the weight of the containing cylinder a minimum, and thus make it possible to arrive at the conditions of the most economic storage and transportation of the acid.

3 All previously published tables of carbonic acid are wholly inadequate for the making of even the simplest engineering calculations involving the economic storage and transportation of the gas. For example, the Report to the British Parliament, in 1896, of the Committee on Compressed Gas Cylinders, generally recognized as containing the most authoritative data relating to the physical properties of carbonic acid, is wholly unreliable in its principal table, page 19, for temperatures exceeding 86 deg. fahr.

4 It is believed that the results of the present investigation make it possible for the first time to state positively the conditions under which the weight of the containing cylinder will be a minimum for

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

the customary conditions of storage and transportation of carbonic acid.

5 In Part I of this paper the tables and charts show the physical properties of pure carbon dioxid and are based upon three things: First, the average of the values obtained by Lord Rayleigh and by Leduc for the weight in grams of one liter of purified and dried carbon dioxid, CO_2, under standard conditions; second, the adjusted results of Amagat's classical experiments showing the precise manner in which carbon dioxid differs in its physical actions from the laws of a perfect gas; and, third, the direct application of certain fundamental physical relations and of mathematical and graphical analyses.

6 In Part II is given the results of the author's experiments on commercial carbonic acid contained in commercial steel cylinders.

7 In Part III is given a rational method of designing commercial carbonic acid cylinders.

PART 1. THE PHYSICAL PROPERTIES OF CARBONIC ACID

8 All the experimental data relating to this subject, so far as available, have been collected and compared. As a proper account of this alone would fill a paper much larger than the present one, only the specific data used as a basis for these new tables and charts will be given. It should be stated, however, that all the data selected for use are of the most reliable character, the original memoirs of the experimenters in every case having been critically examined.

9 All data used have been checked and adjusted by being plotted to a large scale on millimeter coördinate paper, by which means three important clerical errors were discovered and corrected in Amagat's original memoir. Numerous slight adjustments of these data were made in order to have them plot precisely on smooth curves. While making these slight adjustments, great care was taken in every case to have the resulting values follow the apparent law of change.

Weight of Carbonic Acid Gas under Standard Conditions

10 The weight of carbon dioxid, CO_2, known commercially as carbonic acid gas, in grams per liter, under the standard conditions

of one atmospheric pressure and 0 deg. cent. has heretofore been arrived at by two distinctly different methods, giving appreciably different results. These are, first, by the direct or physical method and, second, by the indirect or chemical method.

THE DIRECT OR PHYSICAL METHOD

11 By this method the weight of a certain volume of carbon dioxid is directly compared with the weight of the same volume of a gas of known mass. For the most accurate determination this latter gas has been either air freed from moisture and carbonic acid, or else purified and dried oxygen.

12 The most reliable values thus far obtained for the density of purified and dried carbon dioxid, under standard conditions, are the following:

TABLE A DENSITY OF CARBON DIOXID

Density of carbon dioxid	For air = 1	For oxygen = 1
According to Lord Rayleigh[1]	1.5291	1.3833
According to Leduc[2]	1.5287	1.3832
Average of these values	1.5289	1.3833

13 The most reliable values for the weight in grams of a liter of air, when freed from moisture and carbonic acid, also of purified and dried oxygen, under 760 mm. pressure and 0 deg. cent. are as follows:

TABLE B WEIGHT OF AIR

Weight of air	At Paris	45 deg. latitude and sea level
According to Lord Rayleigh[1]	1.29327	1.29284
According to Leduc[3]	1.29316	1.29273
Average of these values	1.2932	1.2928

[1] Proceedings Royal Society, vol. 62, p. 209, 1897
[2] Comptes Rendus, vol. 126, . p. 415, 1898
[3] Annales de Chimie et de Physique, vol. 15, p. 26, 1898

TABLE C WEIGHT OF OXYGEN

Weight of oxygen	At Paris	45 deg. latitude and sea level
According to Lord Rayleigh[1]	1.42952	1.42905
According to Leduc[2]	1.42867	1.42820
Average of these values	1.4291	1.4286

[1] Proc. Roy. Soc., vol. 62, p. 209, 1897.
[2] Annales de Chimie et de Physique, vol. 15, p. 26, 1898.

14 The weight in grams then of one liter of purified and dried carbon dioxid, at 0 deg. cent., 760 mm. pressure, 45 deg. latitude and sea level,

When referred to air $= 1.5289 \times 1.2928 = 1.9765$.
When referred to oxygen $= 1.3833 \times 1.4286 = 1.9762$.
Average of these values $= 1.976$.

This is the value that has been used in the preparation of the tables and charts contained in this paper, for the weight of carbonic acid gas under standard atmospheric conditions.

15 It should be noted here that all the other experimental data used were taken from Amagat's classical experiments on the physical properties of carbonic acid. See Tables E, F and G.

THE INDIRECT OR CHEMICAL METHOD

16 Most of the standard works containing physical-chemical tables give a value for the weight of carbon dioxid calculated from the atomic weights of its constituents. For example, on page 223 of Physikalisch-Chemische Tablen, Landolt-Börnstein, Berlin, 1905, we find the weight of carbon dioxid, under standard conditions, stated to be 1.965 g. per liter, a value that differs appreciably from 1.976 as above obtained by the direct or physical method. While it is not stated how this value by Landolt-Börnstein was obtained, it was evidently arrived at in the following manner:

17 The molecular weight of carbon dioxid, CO_2, on the basis of the atomic weight of oxygen $= 16$, will of course result from taking the sum of the atomic weight of carbon and twice that of oxygen, which, according to the Table of International Atomic Weights for 1907, will be 44.00. Now by dividing one-half the molecular weight of carbon dioxid by the atomic weight of oxygen we get $22.00 \div 16 =$

1.375; which by this method should be the density of carbon dioxid when that of oxygen is taken as one. This corresponds to 1.3833 as obtained above by the direct or physical method. Now using the same weight as before for a liter of oxygen under standard conditions, we get by the chemical method, in grams per liter, under standard conditions, the weight of carbon dioxid to be $1.375 \times 1.4286 = 1.964$; which corresponds to 1.976 as above obtained by the direct or physical method. Since this value differs by only one milligram from that given by Landolt-Börnstein, it is safe to assume that their value was calculated directly from the molecular weight of the gas.

18 Because of the deservedly great esteem in which this very extensive and excellent set of tables is held by chemists throughout the world, this value has heretofore been practically universally accepted, notwithstanding the fact that preference should have been given to the values obtained by the direct or physical method, as will appear in what follows.

COMPARISON OF THE TWO METHODS

19 A careful study of the direct or physical method has disclosed no inherent reason why there should be any errors in the results obtained other than those due to the purely instrumental and observational inaccuracies.

20 A similar study of the indirect or chemical method, however, has disclosed the fact that this method will of necessity give results that approximate to the truth just in proportion as the gas under consideration approximates to a perfect gas. Now it is well known that carbon dioxid as compared with oxygen, both being under standard conditions, is very much nearer its liquefying state. It must therefore approximate less closely to the state of a perfect gas than does the oxygen whose mass has been used in the calculation of the weight of the carbon dioxid. On the International Atomic Weight basis of oxygen = 16, therefore, carbon dioxid should actually have a somewhat less volume and a correspondingly greater density than that obtained in the customary way by calculation from its molecular weight.

21 That carbon dioxid is in this respect no exception to the general rule followed by other gases having relatively high critical temperatures is evident from an examination of Table D and Fig. A, which show for nine well known gases, whose densities have been accurately determined by direct weighing, that there is a well defined

relation existing between the errors in the calculated densities and the corresponding critical temperatures of these different gases.

TABLE D SHOWING THE RELATION OF THE ERRORS IN THE DENSITIES OF THE WELL KNOWN GASES TO THEIR CRITICAL TEMPERATURES WHEN THE DENSITIES ARE CALCULATED FROM THE MOLECULAR WEIGHTS OF THE GASES

THIS TABLE CONTAINS ONLY THOSE GASES FOR WHICH ACCURATE DETERMINATIONS OF DENSITY HAVE BEEN MADE BY THE METHOD OF DIRECT WEIGHING

Symbol of gas	Molecular weight	Density that of air = 1		Difference in per cent	Critical temperature Degrees centigrade
		Calculated	By direct weighing		
SO_2	64.06	2.2122	2.2639*	2.34	+156
Cl_2	70.90	2.4484	2.4910†	1.74	+141
NH_3	17.03	0.5881	0.5971†	1.53	+131
H_2S	34.08	1.1769	1.1895†	1.07	+100
HCl	36.46	1.2591	1.2692†	0.80	+ 52
CO_2	44.00	1.5194	1.5289‡	0.62	+ 31
NO	30.01	1.0363	1.0387‖	0.23	− 93
CO	28.00	0.9669	0.9672§	0.03	−141
H_2	2.016	0.0696	0.0696°	0.00	−204

* Leduc, Comptes Rendus, Vol. 117, p. 219, 1893
† Leduc, Comptes Rendus, Vol. 125, p. 571, 1897
‡ Average of Rayleigh's and Leduc's values used in this paper
‖ Leduc, Comptes Rendus, Vol. 116, p. 322, 1893
§ Lord Rayleigh, Proceedings Royal Society, Vol. 62, p. 204, 1897
° Lord Rayleigh, Proceedings Royal Society, Vol. 53, p. 134, 1893

22 The critical temperatures given in the last column of this table were taken from Castell-Evans Physico-Chemical Tables, 1902, p. 544. Column 2 of molecular weights is based directly upon the Table of International Atomic Weights for 1907. The densities given in column 3 were obtained by multiplying the corresponding molecular weights by the constant 0.345326, or $\frac{1.4286}{32 \times 1.2928}$ which is the weight of oxygen divided by 32 times the weight of air, both under standard conditions; while the densities given in column 4 were taken directly from the original memoirs of the physicists, as noted. Column 5 shows in per cent the errors resulting from calculating the densities of these gases from their molecular weights.

23 Running the eye down the last two columns of Table D, it will be seen, that when the densities of the different gases are calculated from their molecular weights, the resulting errors in density

are related to the corresponding critical temperatures in such manner that the larger error in calculated density always corresponds to the higher critical temperature. This is brought out very strikingly in Fig. 1, which shows the errors in the calculated densities of these nine well known gases plotted to scale against their critical temperatures. This chart is self-explanatory, and shows conclusively that the method of calculating the density of a gas from its molecular weight is not exact for such gases as have comparatively high critical temperatures.

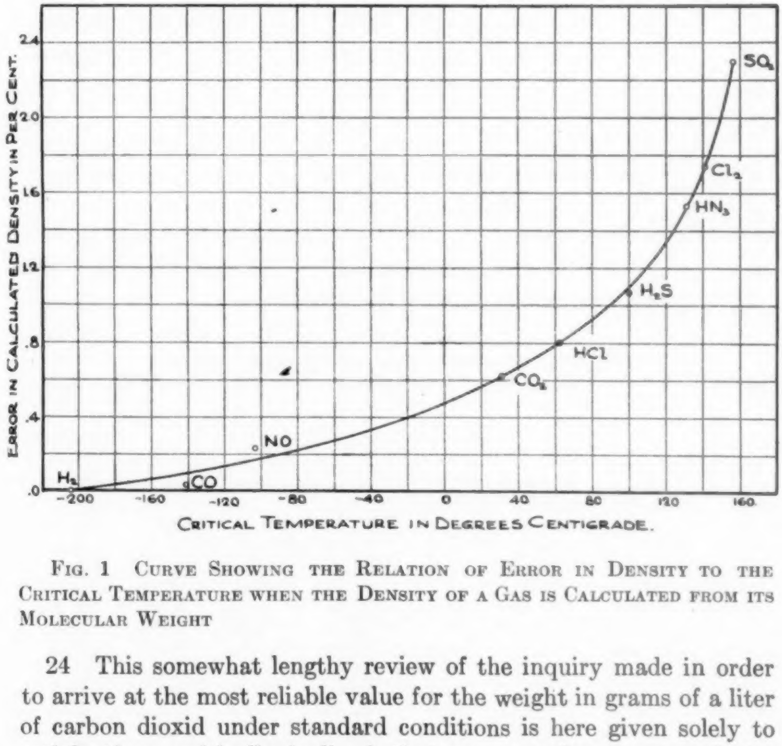

Fig. 1 Curve Showing the Relation of Error in Density to the Critical Temperature when the Density of a Gas is Calculated from its Molecular Weight

24 This somewhat lengthy review of the inquiry made in order to arrive at the most reliable value for the weight in grams of a liter of carbon dioxid under standard conditions is here given solely to satisfy those critically inclined that no error has been made by not using the value commonly found in such standard works as Physikalisch-Chemische Tablen, Landolt-Börnstein, Berlin, 1905.

Amagat's Experimental Researches on Carbonic Acid

25 Emile H. Amagat, the celebrated French physicist, is the author of at least two score memoirs relating to gases. Two of these

contain valuable experimental data on carbon dioxid. These are: Sur la Détermination de la Densité des Gaz et de leur Vapeur Saturée, Comptes Rendus, vol. 114, p. 1093 and p. 1322, 1892; Journal de Physique, p. 288, 1892; Mémoires sur l'Elasticité et la Dilatation des Fluides jusqu'aux trés hautes Pressions, Annales de Chimie et de Physique, vol. 29, 1893.

26 From these two memoirs, the following data were abstracted and form the basis of all the tables contained in Part 1. These data are:

Table E, stating the density of the liquid and of the saturated vapor, also the vapor tension of carbon dioxid in atmospheres, for temperatures ranging from 0 deg. cent. to the critical temperature, 31.35 deg. cent.

Table F, stating the products of the pressures and the corresponding volumes, *PV*, of carbon dioxid, for the temperatures and pressures given.

Table G, supplementary to Table F, stating the values of *PV* at or near which the curves resulting from plotting Table F make abrupt changes in direction.

TABLE E AMAGAT'S DATA FOR CARBON DIOXID

T Degrees Cent.	Density		Maximum vapor tension	T Degrees Cent.	Density		Maximum vapor tension
	Liquid	Saturated vapor			Liquid	Saturated vapor	
0	0.914	0.096	34.3	18.00	0.786	0.176	53.8
1	0.910	0.099	35.2	19.00	0.776	0.183	55.0
2	0.906	0.103	36.1	20.00	0.766	0.190	56.3
3	0.900	0.106	37.0	21.00	0.755	0.199	57.6
4	0.894	0.110	38.0	22.00	0.743	0.208	59.0
5	0.888	0.114	39.0	23.00	0.731	0.217	60.4
6	0.882	0.117	40.0	24.00	0.717	0.228	61.8
7	0.876	0.121	41.0	25.00	0.703	0.240	63.3
8	0.869	0.125	42.0	26.00	0.688	0.252	64.7
9	0.863	0.129	43.1	27.00	0.671	0.266	66.2
10	0.856	0.133	44.2	28.00	0.653	0.282	67.7
11	0.848	0.137	45.3	29.00	0.630	0.303	69.2
12	0.841	0.142	46.4	30.00	0.598	0.334	70.7
13	0.831	0.147	47.5	30.50	0.574	0.356	71.5
14	0.822	0.152	48.7	31.00	0.536	0.392	72.3
15	0.814	0.158	50.0	31.25	0.497	0.422	72.8
16	0.804	0.164	51.2	31.35	0.464	0.464	72.9
17	0.796	0.170	52.4				

TABLE F AMAGAT'S VALUES OF *PV* FOR CARBON DIOXID

P Atmosphere	0 deg. Cent.	10 deg. Cent.	20 deg. Cent.	30 deg. Cent.	40 deg. Cent.	50 deg. Cent.	60 deg. Cent.
1	1.0000	....	....	....	....	....	....
50	0.1050	0.1145	0.6800	0.7750	0.8500	0.9200	0.9840
75	0.1530	0.1630	0.1800	0.2190	0.6200	0.7470	0.8410
100	0.2020	0.2130	0.2285	0.2550	0.3090	0.4910	0.6610
125	0.2490	0.2620	0.2785	0.3000	0.3350	0.3950	0.5100
150	0.2950	0.3090	0.3260	0.3460	0.3770	0.4190	0.4850
175	0.3405	0.3550	0.3725	0.3930	0.4215	0.4570	0.5055
200	0.3850	0.4010	0.4190	0.4400	0.4675	0.5000	0.5425
225	0.4305	0.4455	0.4655	0.4875	0.5130	0.5425	0.5825
250	0.4740	0.4900	0.5100	0.5335	0.5580	0.5865	0.6250
275	0.5170	0.5340	0.5545	0.5775	0.6040	0.6330	0.6675
300	0.5595	0.5775	0.5985	0.6225	0.6485	0.6765	0.7100
350	0.6445	0.6640	0.6850	0.7090	0.7365	0.7650	0.7980
400	0.7280	0.7475	0.7710	0.7950	0.8230	0.8515	0.8840
450	0.8090	0.8310	0.8550	0.8800	0.9075	0.9365	0.9690
500	0.8905	0.9130	0.9380	0.9630	0.9900	1.0210	1.0540
550	0.9700	0.9935	1.0200	1.0465	1.0740	1.1035	1.1370
600	1.0495	1.0730	1.0995	1.1275	1.1570	1.1865	1.2190
650	1.1275	1.1530	1.1800	1.2075	1.2375	1.2680	1.3010
700	1.2055	1.2320	1.2590	1.2890	1.3190	1.3500	1.3825
750	1.2815	1.3105	1.3395	1.3700	1.4000	1.4315	1.4640
800	1.3580	1.3870	1.4170	1.4475	1.4790	1.5105	1.5435
850	1.4340	1.4625	1.4935	1.5245	1.5570	1.5885	1.6225
900	1.5090	1.5385	1.5685	1.6000	1.6325	1.6650	1.6995
950	1.5830	1.6115	1.6430	1.6740	1.7065	1.7395	1.7745
1000	1.6560	1.6850	1.7160	1.7480	1.7800	1.8140	1.8475

P	70 deg.	80 deg.	90 deg.	100 deg.	137 deg.	198 deg.	258 deg.
1	....	....	....	....	....	....	....
50	1.0430	1.0960	1.1530	1.2065	1.3800	....	....
75	0.9180	0.9880	1.0515	1.1180	1.3185	1.6150	1.8670
100	0.7770	0.8725	0.9535	1.0300	1.2590	1.5820	1.8470
125	0.6430	0.7590	0.8580	0.9470	1.2050	1.5530	1.8310
150	0.5750	0.6805	0.7815	0.8780	1.1585	1.5295	1.8180
175	0.5730	0.6515	0.7410	0.8320	1.1230	1.5100	1.8095
200	0.5955	0.6600	0.7315	0.8145	1.0960	1.4960	1.8040
225	0.6285	0.6815	0.7460	0.8175	1.0835	1.4890	1.8035
250	0.6670	0.7135	0.7690	0.8355	1.0810	1.4870	1.8060
275	0.7070	0.7515	0.8015	0.8600	1.0885	1.4875	1.8115
300	0.7485	0.7900	0.8375	0.8900	1.1080	1.4935	1.8200
350	0.8325	0.8725	0.9135	0.9615	1.1565	1.5210	1.8465
400	0.9180	0.9560	0.9960	1.0385	1.2175	1.5630	1.8830
450	1.0035	1.0400	1.0775	1.1190	1.2880	1.6160	1.9280
500	1.0880	1.1240	1.1610	1.2005	1.3620	1.6775	....
550	1.1720	1.2085	1.2430	1.2830	1.4400	1.7450	....
600	1.2540	1.2900	1.3265	1.3655	1.5180	1.8120	....
650	1.3360	1.3725	1.4085	1.4475	1.5960	1.8835	....
700	1.4170	1.4535	1.4900	1.5285	1.6760	1.9560	....
750	1.4985	1.5335	1.5705	1.6100	1.7565	2.0330	....
800	1.5780	1.6140	1.6505	1.6890	1.8355	2.1080	....
850	1.6575	1.6925	1.7285	1.7680	1.9150	2.1860	....
900	1.7345	1.7710	1.8075	1.8460	1.9940	2.2600	....
950	1.8100	1.8470	1.8845	1.9230	2.0720	2.3350	....
1000	1.8840	1.9210	1.9590	1.9990	....	....	....

TABLE G CARBON DIOXID

AMAGAT'S SUPPLEMENTARY TABLE FOR VALUES OF *PV*

P Atmospheres	0 deg. Cent.	10 deg. Cent.	20 deg. Cent.	30 deg. Cent.	32 deg. Cent.	35 deg. Cent.	40 deg. Cent.	50 deg. Cent.	60 deg. Cent.	70 deg. Cent.	80 deg. Cent.	90 deg. Cent.	100 deg. Cent.
31	.7380	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
33	.7120	.7860	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
34	.6990	.7750	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
35	.0750	.7640	.8350	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
37	.0790	.7420	.8170	.8820	.....	.....	.....	.....	.....	.....	.....	.....	.....
40	.....	.7060	.7895	.8590	.8750	.8920	.9235	.....	.....	.....	.....	.....	.....
44	.....	.6530	.7490	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
45	.....	.1050	.7380	.8190	.8350	.8555	.8880	.9520	1.0110	1.0660	.....	.....	.....
48	.....	.....	.7060	.7930	.....	.....	.8670	.9330	0.9950	1.0520	.....	.....	.....
50	.1050	.1145	.6800	.7750	.7920	.8155	.8525	.9210	0.9840	1.0430	1.0980	1.1535	1.2070
53	.....	.....	.6370	.7460	.....	.....	.8300	.9020	0.9680	1.0280	1.0850	1.1420	1.1960
55	.....	.....	.6050	.7260	.7455	.7720	.8135	.8890	0.9570	1.0185	1.0760	1.1340	1.1885
56	.....	.....	.5850	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
57	.....	.....	.1480	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
60	.....	.....	.1520	.6680	.6935	.7245	.7720	.8555	0.9285	0.9940	1.0540	1.1130	1.1710
65	.....	.....	.....	.5950	.6290	.6690	.7260	.8200	0.8990	0.9690	1.0325	1.0930	1.1530
68	.....	.....	.....	.5350	.5780	.6310	.6950	.7970	0.8810	0.9530	1.0190	1.0810	1.1420
70	.....	.....	.....	.4700	.5400	.6020	.6730	.7820	0.8685	0.9430	1.0100	1.0730	1.1350
71	.....	.....	.....	.2300	.....	.....	.....	.....	.....	.....	.....	.....	.....
72	.....	.....	.....	.2230	.4910	.....	.....	.....	.....	.....	.....	.....	.....
73	.....	.....	.....	.....	.4600	.....	.....	.....	.....	.....	.....	.....	.....
74	.....	.....	.....	.2190	.4050	.5310	.....	.....	.....	.....	.....	.....	.....
74.5	.....	.....	.....	.....	.3400	.....	.....	.....	.....	.....	.....	.....	.....
75	.....	.....	.....	.2190	.2680	.5100	.6130	.7410	0.8360	0.9170	0.9880	1.0535	1.1180
76	.....	.....	.....	.....	.2495	.4850	.....	.....	.....	.....	.....	.....	.....
78	.....	.....	.....	.2205	.2410	.4200	.....	.....	.....	.....	.....	.....	.....
80	.....	.....	.....	.2225	.....	.3180	.5400	.7000	0.8030	0.8900	0.9660	1.0335	1.1005
82	.....	.....	.....	.....	.....	.2810	.5030	.....	.....	.....	.....	.....	.....
85	.....	.....	.....	.....	.....	.2670	.4350	.6510	0.7690	0.8630	0.9425	1.0135	1.0835
90	.....	.....	.....	.....	.....	.2650	.3410	.5990	0.7340	0.8350	0.9190	0.9935	1.0655
95	.....	.....	.....	.....	.....	.....	.3140	.5460	0.6980	0.8060	0.8960	0.9735	1.0480
100	.....	.....	.....	.....	.....	.....	.3090	.4910	0.6610	0.7770	0.8720	0.9540	1.0305
110	.....	.....	.....	.....	.....	.....	.3130	.4170	0.5880	0.7210	0.8240	0.9140	0.9970

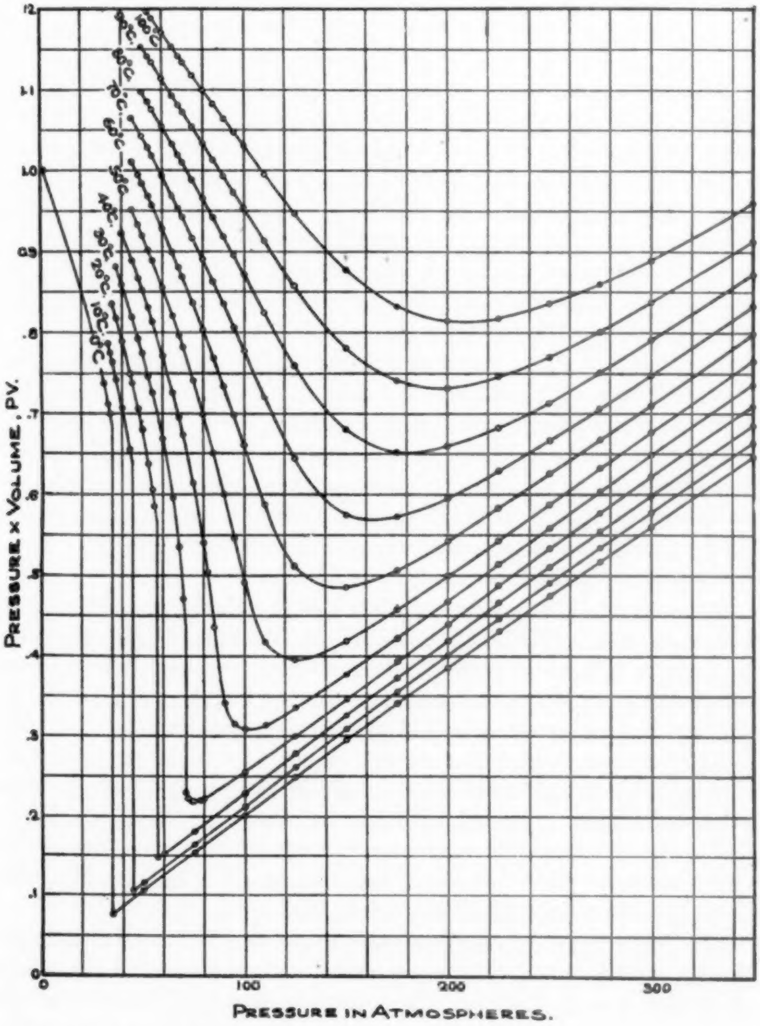

FIG. 2 CURVES SHOWING THE RELATION OF AMAGAT'S *PV* VALUES TO THE PRESSURE OF CARBONIC ACID

Tables of the Physical Properties of Carbonic Acid, as Calculated by the Author in 1904 and Recalculated under his Direction by Frank P. Kramer in 1907

27 Table 1 shows, in metric units, the physical properties of the liquid and of the saturated vapor of carbon dioxid, for temperatures ranging from 0 deg. cent. to the critical temperature, 31.35 deg. cent.

28 In this table the liquid and the vapor are assumed to be in the condition in which they exist in a cylinder when partially filled with liquid carbon dioxid, the space overlying the liquid being filled with the saturated vapor. A full explanation of the critical temperature and of these states will be found elsewhere in this paper under the caption, The Three States of Carbonic Acid.

29 Columns 2, 4 and 5 of this table are based directly upon Amagat's experimental results, Table E, the only difference being that the slight irregularities of Amagat's values have been carefully adjusted in Table 1. This was accomplished by plotting his results to a large scale on millimeter coördinate paper, after which the plotted values were slightly shifted, where necessary, in order to make them plot precisely on smooth curves, and thus bring them into closer agreement with the apparent laws involved.

30 Column 3 was then calculated from column 2, by means of the conversion factor 1.03329, which is the fluid pressure in kilograms per square centimeter corresponding to one atmosphere. This factor was taken from Hering's Conversion Tables, vol. 1, p. 66, 1904, and was checked before using, by independent calculation.

31 Columns 6 and 7 were calculated from columns 4 and 5, respectively, by taking the reciprocals of the corresponding entries in these columns. It will be observed that this simple relation is due to the fact that a liter of water under standard conditions weighs one kilogram. That is to say, the volumes in liters per kilogram should be the reciprocals of the corresponding densities.

32 After these calculations were made and entered in the preliminary table, each column so calculated was checked by being plotted to a large scale on millimeter coördinate paper; after which the final figure, where found necessary, was adjusted so as to bring the plotted result precisely on the curve representing the apparent laws involved. This final adjustment was necessary only when for obvious reasons the reciprocal had to be stated with somewhat greater precision than that of the constant from which it was obtained.

33 Table 2 shows in British units the physical properties of the liquid and of the saturated vapor of carbon dioxid, for temperatures ranging from 32 deg. fahr. to the critical temperature, 88.4 deg. fahr. For explanation of the critical temperature, etc., see elsewhere in this paper, The Three States of Carbonic Acid.

34 This table was prepared from Table 1 in the following manner: First, columns 2, 4 and 5 of Table 1 were plotted to large scales on millimeter coördinate paper, the vertical scale of each being centigrade degrees; second, on each of these charts a second vertical scale of equivalent fahrenheit degrees was then constructed; and third, from curves adjusted to the values thus plotted to centigrade degrees there were then read the equivalent values from the fahrenheit scales.

35 This graphical method of conversion from one to the other temperature scale worked very satisfactorily indeed. It of course gives theoretically correct results, even when, as in this case, the equations of the curves involved are unknown. In this manner columns 3, 4 and 5 of Table 2 were obtained, showing in British units the fundamental relations of temperature, pressure, and density for both the liquid and the saturated vapor of carbon dioxid, for temperatures less than the critical temperature, 88.4 deg. fahr.

36 Column 2 was then calculated from column 3 by use of the conversion factor 14.697, which is the equivalent in pounds of one atmospheric pressure. Columns 6 and 8 were calculated from 4 by use of the conversion factors 62.43 and 8.345, which are the respective weights in pounds of 1 cu. ft. and 1 U. S. gal. of pure water at its maximum density; and columns 7 and 9 were similarly calculated from 5. Finally columns 10 and 11 were calculated respectively from 6 and 7, as in Table 1, by taking the reciprocals of the corresponding constants in those columns.

37 After all these calculations were made, and before the results were finally entered in the table, each column, as in Table 1, was adjusted to plot precisely on smooth curves.

38 Table 3 shows the pressures of carbon dioxid, in pounds per square inch, corresponding to densities ranging from 0.50 to 1.00, that of water at its maximum density being unity, and for temperatures ranging from 0 deg. to 100 deg. cent. It was calculated directly from Amagat's PV values as follows:

39 First, a table of densities corresponding to each PV value of Tables F and G was calculated by means of the formula

$$G = \frac{G_0 P}{C} = \frac{0.001976P}{C} \qquad (1)$$

where

G = the density of carbon dioxid corresponding to any tabular pressure, that of water at 4 deg. cent. being unity.

G_0 = 0.001976 = the density of carbon dioxid at 760 mm., 0 deg. cent., 45 deg. latitude, and sea level.

P = the pressure exerted by the carbon dioxid in standard atmospheres.

C = the tabular value of PV as read from Amagat's Tables F and G.

40 Second, these calculated values were then plotted in a manner similar to Fig. 2, but to a large scale on millimeter coördinate paper, the horizontal scale representing the densities referred to water and the vertical scale the corresponding pressures in pounds per square inch; the atmospheric pressures, before plotting, having been reduced to the equivalent pressures in pounds per square inch. Smooth curves were then adjusted to these plotted values, after which the entries of Table 3 were obtained by direct readings from these adjusted curves.

DERIVATION OF FORMULA ONE

41 This formula is based upon the following considerations:—

42 First, the density of a fluid referred to water will be the same as its weight in kilograms per liter. The density of carbon dioxid therefore, when referred to water at 4 deg. cent. will be 0.001976 under standard conditions, since a liter under these conditions weighs 1.976 g. For the derivation of this value see, elsewhere in this paper, the discussion of Lord Rayleigh's and Leduc's experimental determinations relating to carbon dioxid.

43 Second, Amagat's Tables F and G give for the different temperatures and pressures of carbon dioxid the results obtained by multiplying each pressure by its corresponding volume, the result being expressed in terms of this product under the standard conditions of one atmospheric pressure and 0 deg. cent. That is to say,

$$PV = CP_0V_0 = CV_0$$

where P_0 represents one atmospheric pressure and V_0 the corresponding volume at 0 deg. cent. From this we get the ratio of volume

$$\frac{V_0}{V} = \frac{P}{C}$$

Since the density of a definite mass of gas is always inversely as the volume that it occupies, we get

$$\frac{G}{G_o} = \frac{V_o}{V} = \frac{P}{C}$$

or

$$G = G_o \frac{P}{C} = W \frac{P}{C}$$

where W is the weight in kilograms per liter under the standard conditions of one atmospheric pressure, 0 deg. cent., 45 deg. latitude, and sea level; which for purified and dried carbon dioxid is 0.001976.

44 Table 4 gives the pressures of carbon dioxid corresponding to densities ranging from 0.50 to 1.00, that of water at its maximum density being unity, and for temperatures ranging from 32 deg. to 212 deg. fahr. It should be noted, for that portion of the table corresponding to temperatures less than the critical temperature of 88.4 deg. fahr. and lying above the line AB, that the total weight of carbon dioxid contained in a cylinder is made up of two parts, one of which is the liquid portion and the other the vaporous portion which always fills the volume of the cylinder not occupied by the liquid. For this portion of the table the combined density of the liquid and its overlying vapor is here tabulated. A further explanation of this part of the table will be found elsewhere in this paper under the caption, Combined Density of the Liquid and Saturated Vapor.

45 This table was obtained as follows: First, a chart, constructed to a large scale on millimeter coördinate paper, the horizontal scale representing pressures in pounds per square inch, and the vertical scale temperatures in degrees centigrade, was made from readings taken from the large scale chart, similar to Fig. 2, that was used in the preparation of Table 3. Second, a fahrenheit temperature scale was interpolated on this chart, after which the pressures and corresponding densities for the different fahrenheit temperatures were read and plotted to a large scale in a manner similar to Fig. 2, the temperature curves in this case being, however, fahrenheit instead of centigrade degrees. Third, the entries of this table were then obtained by direct readings from the fahrenheit temperature curves of this final chart.

46 This proved a most satisfactory method for obtaining the values of pressures in pounds corresponding to given densities and

fahrenheit temperatures, from Amagat's *PV* values corresponding to pressures in atmospheres and centigrade temperatures. This method when carried out to the large scales used, gives results that are fully as precise as the accuracy of Amagat's experimental results warrant, and in addition, like other similar graphical methods, is practically self-checking.

47 Table 5 shows, for different temperatures and combined densities, or proportions of filling in pounds of carbonic acid per pound of water capacity, the portion of the cylinder volume occupied by the liquid part of the carbonic acid contained.

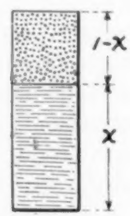

FIG. 3

48 The total weight of carbonic acid charged into a cylinder, when below the critical temperature of 88.4 deg. fahr., will ordinarily be made up of a liquid portion and of a portion that is in the state of a saturated vapor, which overlies the liquid and fills the volume of the cylinder unoccupied by it.

49 This table was calculated by means of the following formula, derived as follows:

Let

x = the portion of the cylinder volume that is occupied by the liquid portion of the carbonic acid.

$1-x$ = the portion occupied by the saturated vapor.

G_l = the density of the liquid carbonic acid, that of pure water at 4 deg. cent. being unity.

G_v = ditto of the saturated vapor of carbonic acid.

G = combined density of the liquid and saturated vapor of carbonic acid, or the proportion of filling in pounds of carbonic acid per pound of water capacity.

Then the volume of the liquid portion x multiplied by its density G_l, or $x\,G_l$, added to the volume of the saturated vapor $1-x$, multiplied by its density G_v, or $(1-x)G_v$,

must equal the volume of the cylinder 1, multiplied by the combined density G; or

$$x\, G_l + (1-x)\, G_v = G$$

or

$$x = \frac{G - G_v}{G_l - G_v} \tag{2}$$

That is, the portion of the cylinder volume occupied by the liquid part of the carbonic acid contained is equal to the difference of the combined density and that of the saturated vapor, divided by the difference of the density of the liquid and that of the saturated vapor.

50 When applying this formula it should be remembered that the liquid density here involved is that given in column 4 of Tables 1 and 2; and is for the liquid when in contact with its saturated vapor, that is to say, when it is just on the point of being converted into a vapor. The saturated vapor densities to be used are given in column 5 of the same tables.

The Three States of Carbonic Acid

51 Carbonic acid as commercially handled may exist in three distinctly different states: the gaseous, the vaporous, and the liquid. Other states than these have no bearing upon questions of commercial storage and transportation, and will not therefore be considered in this paper.

52 Commercial carbonic acid is manufactured, collected as a by-product, or from natural sources, as either a gas or a superheated vapor. After being compressed and cooled sufficiently this gas or vapor is converted into a liquid, which is charged into steel cylinders for transportation to the consumer.

53 From the standpoint of the physicist the critical temperature is that temperature above which a substance always exists in the gaseous state. For temperatures above the critical temperature no substance has yet been reduced to the liquid state by any pressure, however great; while for temperatures below the critical temperature all the commonly occurring gases have been liquefied.

54 The critical temperature of purified and dried carbon dioxid, as determined by Amagat, is 31.35 deg. cent., or 88.4 deg. fahr. He

has compressed the gas while above this temperature to a pressure approximating 15 000 lb. per square inch, and thus to a density about 10 per cent greater than that of water, without reducing it to a liquid. For any temperature below this critical temperature he found a definite and fixed pressure at which the carbon dioxid could be reduced to the liquid state. The pressures at which liquefaction occurs for the different temperatures are given in columns 2 and 3 of Table 1, in metric units, and in the same columns of Table 2, in British units.

GASEOUS, SATURATED AND LIQUID STATES

55 For all temperatures above the critical temperature of 31.35 deg. cent., or 88.4 deg. fahr., carbon dioxid is always found in the gaseous state. While in this state it necessarily exists as a homogeneous substance that fills all parts of the containing vessel, just as would the more perfect gases, oxygen and hydrogen. While in this state it cannot be reduced to a liquid by any pressure, however great.

56 The relations of density, temperature and pressure of purified and dried gaseous carbon dioxid are given in the columns corresponding to temperatures from 31.35 to 100 deg. cent., inclusive, of Table 3, and to temperatures from 88.4 to 212 deg. fahr. inclusive, of Table 4.

57 Carbon dioxid exists as a saturated vapor when just on the point of being converted into a liquid. It can exist in this state only when its temperature is less than the critical temperature of 31.35 deg. cent., or 88.4 deg. fahr. When in this state, just as in the case of saturated steam, for every pressure there is a corresponding definite temperature and density. These relations of temperature, pressure and density are given in Metric units in Table 1, and in British units in Table 2.

58 Carbon dioxid can exist in the liquid state, first, only for temperatures less than the critical temperature of 31.35 deg. cent., or 88.4 deg. fahr., and, second, for any such temperature, only for pressures that are equal to or greater than the saturated vapor pressure corresponding to that temperature.

59 There are two entirely distinct conditions under which liquid carbon dioxid may be stored for transportation. It is extremely important that these two conditions should be clearly understood. They are, first, when the liquid carbon dioxid partially fills the containing cylinder, and second, when it entirely fills the containing cylinder. The importance of differentiating these two conditions is apparent when it is considered that the pressures exerted on the

walls of the cylinders under these two conditions m^ay be widely different for any given temperature, as will appear from what follows.

LIQUID STATE FOR PARTIAL FILLING

60 For this case the relation of temperature, pressure, and density of purified liquid carbon dioxid are given in metric units in columns 1 to 4 inclusive of Table 1, and in British units in the same columns of Table 2.

61 These two tables state the properties of liquid carbon dioxid when in the condition of just having been condensed from the vaporous state into a liquid, or just on the point of being converted from a liquid into the vaporous state. This is always the state of the flush liquid portion of the contents of a cylinder when the liquid does not entirely fill it, the space unoccupied by the liquid being of course filled with a saturated vapor.

LIQUID STATE FOR COMPLETE FILLING

62 When the pressure on the liquid is greater than that exerted by the saturated vapor at the same temperature it is evident that no saturated vapor can exist in contact with the liquid, so that for this condition the liquid carbon dioxid must entirely fill the containing cylinder.

63 The relations of density, temperature, and pressure for this condition are those shown as lying below the line *AB* of Tables 3 and 4. For example, in Table 4, for a temperature of 70 deg. fahr. the pressures corresponding to densities of 0.80, 0.90, and 1.00 are respectively 1060, 2120, and 4940 lb. per square inch; while for the same temperature for the condition of partial liquid filling of the cylinder, corresponding to combined densities of the liquid and the overlying saturated vapor ranging from 0.50 to 0.75, the pressure is constant, and equals 849 lb. per square inch.

COMBINED DENSITY OF THE LIQUID AND SATURATED VAPOR

64 In cylinders under the ordinary conditions of storage and transportation, for temperatures less than the critical temperature of 31.35 deg. cent., or 88.4 deg. fahr., we have to deal with the carbonic acid in both the liquid and the vaporous states at the same time.

That is to say, of the total weight of carbonic acid charged into a cylinder a portion is in the liquid state while the remainder exists in the state of a saturated vapor filling the volume of the cylinder that is unoccupied by the liquid portion. In the commercial storage of carbonic acid we are therefore obliged to deal with the combined density of the liquid and its overlying vapor. The portions of the cylinder volume that are occupied by the liquid part of the carbonic acid charged into a cylinder have been worked out for combined densities ranging from 0.50 to 1.00 and are given in Table 5.

65 The portions of Tables 3 and 4 above the line AB show the relations of the density, temperature and pressure of carbon dioxid for the condition of partial liquid filling of the cylinder. The densities in column 1 for this portion of each of these tables are the combined densities of the liquid and its overlying vapor, for the different conditions tabulated; and are equal, in each case, to their combined weights divided by the weight of their combined volumes of water, the water being at its maximum density, or at a temperature of 4 deg. cent., or 39.2 deg. fahr. In other words, the densities here given are the same as the corresponding "proportion of filling in pounds of carbonic acid per pound of water capacity." That is to say, a tabular density of 0.62 means, for any portion of these tables, that the weight of carbonic acid contained is 62 per cent of the weight of the water that would be required to fill the cylinder entirely, the water being pure and at its maximum density.

66 It will be observed that for this portion of these tables, namely, that lying above the line AB, there is a fixed relation existing between the temperature and pressure irrespective of the value of the combined density. For example, for a temperature of 60 deg. fahr., Table 4, the pressure is 744 lb. per square inch for all combined densities of from 0.50 to 0.80 inclusive. This means for this temperature and range of density, that the cylinder contains both liquid carbon dioxid and its saturated vapor; the portion of the cylinder volume occupied by the liquid increasing, and that occupied by the saturated vapor diminishing, as the combined density increases.

67 Now so long as there is vapor present, that is, so long as the liquid does not completely fill the cylinder, the pressure exerted will of course be that due to the saturated vapor alone, and is therefore constant for any fixed temperature; and of course entirely independent of the portion of the cylinder volume that is filled with the liquid.

68 When the cylinder, however, becomes entirely filled with the

liquid, so that no space remains for vapor, then the pressure will cease to be constant for a given temperature. By referring to Table 4, this is seen to occur for a temperature of 50 deg. fahr. between the densities of 0.85 and 0.86; for 60 deg. fahr., between those of 0.80 and 0.81; for 70 deg. fahr., between those of 0.75 and 0.76; and for 80 deg. fahr., between those of 0.67 and 0.68.

PORTION OF CYLINDER VOLUME OCCUPIED BY THE LIQUID

69 In the commercial storage and transportation of carbonic acid, for temperatures less than the critical temperature of 88.4 deg. fahr. it is a matter of interest to know just what portions of the cylinder volume are occupied by the liquid and by the saturated vapor. An investigation of this matter has resulted in the production of Table 5; which shows for different temperatures and combined densities, which latter is of course the same thing as the "proportion of filling in pounds of carbonic acid per pound of water capacity," the precise portion of the cylinder volume that is occupied by the liquid portion of the carbonic acid contained. For example, if a cylinder be charged, at a temperature of 60 deg. fahr. to 0.62 of its water capacity by weight, then, from Table 5, we find that the resulting liquid part of the carbonic acid will occupy 0.708 of the cylinder volume, the remaining portion, or 0.292, being occupied by the saturated vapor. If now, the temperature of the cylinder and its contents, for the same proportion of filling, be elevated to 80 deg. fahr., then the liquid will occupy according to this table, 0.863 of the cylinder volume. This increase in the volume of the liquid portion of the cylinder contents is due, first, to the expansion of the original liquid portion under the influence of the rise in temperature, and second, to the condensation to the liquid state of a portion of the original saturated vapor. From the table, this action is seen to continue, as the temperature rises, until a temperature somewhere between 84 and 86 deg. fahr. is reached, at which point all the vapor is condensed, and the whole of the carbonic acid content is in the liquid form, and of course fills the entire cylinder volume.

70 After the completion of Table 5 it was discovered that a sealed glass tube containing liquid carbonic acid, procured from Germany for experimental purposes, did not follow the law of increase of liquid volume with rise of temperature as exhibited by this table. On the contrary, as the temperature of the tube was elevated the volume of the liquid gradually decreased and altogether vanished before the

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70 After the completion of Table 5 it was discovered that a sealed glass tube containing liquid carbonic acid, procured from Germany for experimental purposes, did not follow the law of increase of liquid volume with rise of temperature as exhibited by this table. On the contrary, as the temperature of the tube was elevated the volume of the liquid gradually decreased and altogether vanished before the

critical temperature was reached. This unexpected action seemed inexplicable at first, but an investigation revealed the curious law that is apparent from an inspection of Fig. 3, which has been constructed for a much larger range in combined density than Table 5.

71 An inspection of Fig. 3 shows that for combined densities greater than the critical density of 0.46 an increase in temperature is always attended by an increase in the liquid volume, while for densities less than 0.46 the law is reversed, so that any increase in temperature is attended by a decrease in liquid volume. Upon making a determination for the combined density of the carbonic acid in the sealed glass tube, which was found to be 0.3, that of Table 5 being 0.5 to 1.0, the matter became quite clear when viewed in the light of the law revealed by Fig. 3.

PRESSURES CORRESPONDING TO DIFFERENT DENSITIES OR PROPORTIONS OF FILLING, IN POUNDS OF CARBONIC ACID PER POUND OF WATER CAPACITY

72 These pressures may be read in metric units, directly from Table 3, and in British units from Table 4, for a large range in both temperature and density. The densities here given corresponding to pressures lying above the line *AB* are the combined densities, as fully explained elsewhere in this paper under the caption "Combined densities of the liquid and saturated vapor." The densities given in column 1 are for all parts of the table the same as the "proportion of filling in pounds of carbonic acid per pound of water capacity." Thus for a cylinder charged with carbonic acid so as to contain 0.62 of the weight of its capacity in water, or a density of 0.62, the pressures corresponding to temperatures of 50, 80 and 110 deg. fahr. are respectively 650, 965, and 1560 pounds per square inch; while those corresponding to the same temperatures for a cylinder charged to 0.84 of its water capacity would be respectively 650, 1780 and 3115 pounds. This shows that for these two proportions of filling, while at 50 deg fahr. the pressures are precisely the same, the cylinder containing the greater proportion of filling will be subjected to a pressure that is 84 per cent greater at 80 deg. fahr., and that is 99 per cent greater at 110 deg. fahr.

TABLE 1 PHYSICAL PROPERTIES OF THE LIQUID AND THE SATURATED VAPOR OF CARBONIC ACID FROM 0 DEG. CENT. TO THE CRITICAL TEMPERATURE 31.35 DEG. CENT.

PRESSURE FACTORS.–For pounds per square inch multiply by 14.697
for Column 2

WEIGHT FACTORS for Columns 4 & 5.

For grams per cubic centimeter	multiply by	1.
" kilograms per cubic meter	" "	1000.
" kilograms per liter	" "	1.
" pounds per cubic inch	" "	0.03613
" pounds per cubic foot	" "	62.43
" pounds per gallon (U.S.)	" "	8.345

VOLUME FACTORS for Columns 6 & 7

For cubic centimeters per gram	multiply by	1.
" cubic meters per kilogram	" "	0.001
" cubic inches per pound	" "	27.68
" cubic feet per pound	" "	0.01602
" gallons (U.S.) per pound	" "	0.1198

Temperature Degrees Centigrade	Pressure, Saturated Vapor		Density, That of Water at 4°C.=1		Volume, Liters per Kilogram	
	Atmospheres	Kilograms per Sq. Cent.	Liquid	Vapor	Liquid	Vapor
0	34.3	35.4	.916	.096	1.092	10.42
1	35.2	36.35	.911	.099	1.098	10.09
2	36.1	37.3	.906	.1025	1.104	9.76
3	37.05	38.3	.900	.106	1.111	9.44
4	38.0	39.3	.894	.1095	1.118	9.13
5	39.0	40.3	.888	.113	1.126	8.84
6	40.0	41.35	.882	.117	1.134	8.55
7	41.0	42.4	.876	.121	1.142	8.27
8	42.05	43.45	.869	.125	1.151	8.00
9	43.1	44.55	.862	.129	1.160	7.75
10	44.2	45.65	.855	.133	1.170	7.51
11	45.3	46.8	.848	.1375	1.180	7.27
12	46.45	48.0	840	.142	1.191	7.03
13	47.6	49.2	.832	.147	1.202	6.80
14	48.8	50.4	.823	.152	1.215	6.57
15	50.0	51.65	.814	.158	1.228	6.34
16	51.2	52.9	.805	.164	1.242	6.12
17	52.4	54.2	.796	.170	1.256	5.90
18	53.7	55.5	.786	.176	1.272	5.68
19	55.0	56.8	.776	.183	1.289	5.46
20	56.3	58.15	.765	.191	1.307	5.24
21	57.6	59.5	.754	.199	1.326	5.03
22	59.0	60.95	.743	.208	1.346	4.81
23	60.4	62.4	.731	.217	1.368	4.60
24	61.8	63.85	.718	.228	1.393	4.39
25	63.2	65.3	.703	.240	1.422	4.18
26	64.7	66.85	.688	.252	1.454	3.97
27	66.2	68.4	.671	.266	1.490	3.76
28	67.7	69.95	.653	.282	1.531	3.54
29	69.2	71.5	.630	.303	1.587	3.30
30	70.7	73.1	.598	.334	1.672	2.99
30.5	71.5	73.9	.574	.356	1.742	2.80
31	72.3	74.7	.536	.392	1.866	2.55
31.25	72.7	75.2	.497	.422	2.012	2.37
31.35	72.9	75.35	.464	.464	2.155	2.155

TABLE 2 PHYSICAL PROPERTIES OF THE LIQUID AND THE SATURATED VAPOR OF CARBONIC ACID FROM 32 DEG. FAHR. TO THE CRITICAL TEMPERATURE 88.4 DEG. FAHR.

Temp. Deg. Fah	Pressure, Saturated Vapor		Density, Water at 39.2°F.=1		Wt. of One Cubic Ft. Pounds		Wt. of One Gallon Pounds		Volume of One Lb. Cubic Feet	
	Lbs. per Sq. In.	Atmospheres	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor	Liquid	Vapor
32	504	34.3	.916	.096	57.2	5.99	7.64	.801	.0175	.167
33	511	34.8	.913	.098	57.0	6.10	7.62	.816	.0175	.164
34	519	35.3	.910	.100	56.8	6.21	7.59	.831	.0176	.161
35	526	35.8	.907	.101	56.6	6.32	7.57	.846	.0177	.158
36	534	36.3	.904	.103	56.4	6.44	7.54	.861	.0177	.155
37	542	36.9	.901	.105	56.2	6.55	7.52	.877	.0178	.152
38	549	37.4	.898	.107	56.1	6.67	7.49	.893	.0178	.149
39	557	37.9	.895	.109	55.9	6.80	7.47	.910	.0179	.147
40	565	38.5	.892	.111	55.7	6.92	7.44	.927	.0180	.144
41	573	39.0	.888	.113	55.5	7.05	7.42	.944	.0180	.142
42	581	39.5	.885	.115	55.3	7.18	7.39	.961	.0181	.139
43	589	40.1	.882	.117	55.1	7.31	7.36	.979	.0182	.136
44	598	40.7	.878	.120	54.8	7.45	7.33	.996	.0183	.134
45	606	41.2	.875	.122	54.6	7.59	7.30	1.015	.0183	.132
46	614	41.8	.871	.124	54.4	7.73	7.27	1.033	.0184	.129
47	623	42.4	.867	.126	54.1	7.87	7.24	1.05	.0185	.127
48	632	43.0	.863	.128	53.9	8.01	7.20	1.07	.0186	.125
49	641	43.6	.859	.131	53.6	8.16	7.17	1.09	.0187	.122
50	650	44.2	.855	.133	53.4	8.31	7.13	1.11	.0187	.120
51	659	44.8	.851	.136	53.1	8.46	7.10	1.13	.0188	.118
52	668	45.5	.847	.138	52.9	8.62	7.07	1.15	.0189	.116
53	677	46.1	.843	.141	52.6	8.79	7.03	1.17	.0190	.114
54	686	46.7	.838	.143	52.3	8.95	6.99	1.20	.0191	.112

TABLE—2 CONTINUED.

55	696	47.4	.833	.146	52.0	9.12	6.95	1.22	.0192	.110
56	705	48.0	.829	.149	51.7	9.30	6.92	1.24	.0193	.108
57	715	48.6	.824	.152	51.4	9.48	6.88	1.27	.0194	.106
58	725	49.3	.819	.155	51.1	9.67	6.83	1.29	.0196	.104
59	735	50.0	.814	.158	50.8	9.86	6.79	1.32	.0197	.102
60	744	50.7	.809	.161	50.5	10.1	6.75	1.34	.0198	.0999
61	754	51.3	.804	.164	50.2	10.3	6.71	1.37	.0199	.0978
62	764	52.0	.799	.168	49.9	10.5	6.67	1.40	.0200	.0958
63	775	52.7	.794	.171	49.6	10.7	6.63	1.43	.0202	.0938
64	785	53.4	788	.175	49.2	10.9	6.58	1.46	.0203	.0918
65	796	54.1	.783	.178	48.9	11.1	6.53	1.49	.0205	.0898
66	806	54.8	.777	.182	48.5	11.4	6.48	1.52	.0206	.0879
67	817	55.6	.771	.186	48.1	11.6	6.43	1.55	.0208	.0859
68	827	56.3	.765	.191	47.8	11.9	6.38	1.59	.0209	.0840
69	838	57.0	.759	.195	47.4	12.2	6.33	1.63	.0211	.0821
70	849	57.8	.753	.200	47.0	12.5	6.28	1.67	.0213	.0802
71	860	58.5	.747	.205	46.6	12.8	6.23	1.71	.0214	.0783
72	872	59.3	.740	.210	46.2	13.1	6.18	1.75	.0216	.0764
73	883	60.1	.733	.215	45.8	13.4	6.12	1.79	.0218	.0745
74	894	60.8	.726	.221	45.4	13.8	6.06	1.84	.0220	.0726
75	906	61.6	.719	.227	44.9	14.2	6.00	1.89	.0223	.0707
76	917	62.4	.711	.233	44.4	14.6	5.93	1.94	.0225	.0689
77	929	63.2	.703	.240	43.9	15.0	5.87	2.00	.0228	.0670
78	941	64.0	.695	.246	43.4	15.4	5.80	2.05	.0231	.0651
79	953	64.9	.686	.253	42.8	15.8	5.72	2.11	.0234	.0633
80	965	65.7	.677	.261	42.2	16.3	5.65	2.18	.0237	.0614
81	977	66.5	.667	.269	41.6	16.8	5.57	2.25	.0240	.0595
82	989	67.3	.657	.278	41.0	17.4	5.48	2.32	.0244	.0575
83	1002	68.2	.646	.288	40.3	18.0	5.39	2.40	.0248	.0555
84	1015	69.0	.633	.300	39.5	18.7	5.28	2.50	.0253	.0533
85	1027	69.9	.618	.315	38.6	19.7	5.16	2.63	.0259	.0508
86	1039	70.7	.598	.334	37.3	20.9	4.99	2.79	.0268	.0479
87	1052	71.6	.571	.360	35.6	22.5	4.76	3.00	.0281	.0445
88	1065	72.5	.523	.404	32.7	25.2	4.36	3.37	.0308	.0397
88.4	1071	72.9	.464	.464	29.0	29.0	3.87	3.87	.0345	.0345

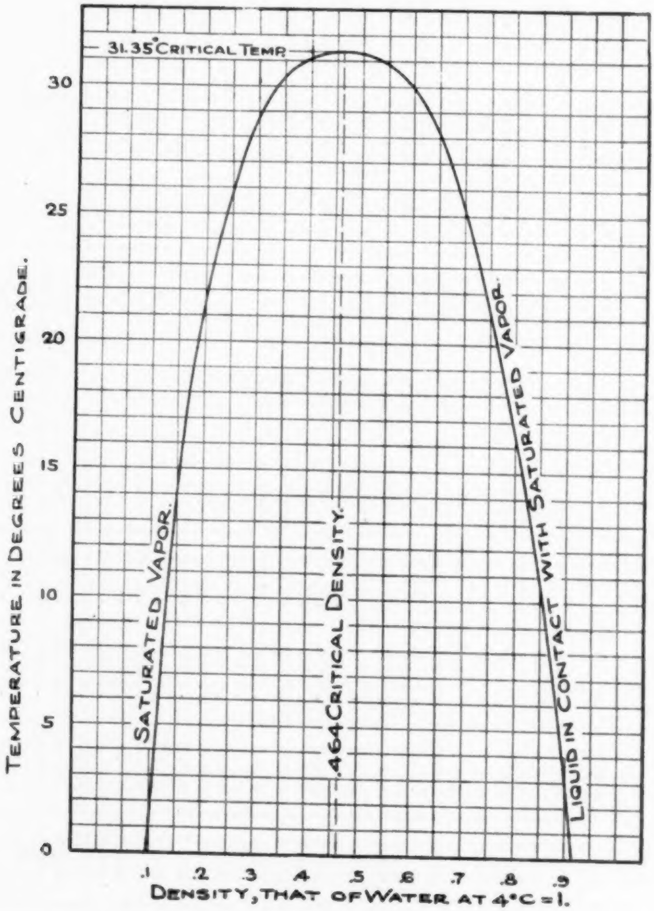

Fig. 1 Curve Showing the Relation of Density to Temperature of Carbonic Acid for Temperatures less than Critical Temperature

TABLE 3 PRESSURE EXERTED BY CARBONIC ACID ON THE WALLS OF THE CONTAINING VESSEL, IN POUNDS PER SQUARE INCH

PORTION OF TABLE ABOVE *AB* IS FOR COMBINED DENSITY OF LIQUID AND OVERLYING VAPOR

Density, Water at 4°C=1	Temperature, Degrees Centigrade.										
	0°	10°	20°	30°	40°	50°	60°	70°	80°	90°	100°
.50	504	650	827	1039	1305	1585	1870	2155	2445	2730	3020
.51	504	650	827	1039	1310	1600	1895	2190	2485	2780	3080
.52	504	650	827	1039	1320	1615	1915	2220	2525	2830	3140
.53	504	650	827	1039	1330	1630	1940	2255	2570	2880	3205
.54	504	650	827	1039	1335	1650	1970	2295	2615	2940	3270
.55	504	650	827	1039	1345	1670	1995	2330	2665	3000	3340
.56	504	650	827	1039	1355	1690	2025	2370	2715	3060	3410
.57	504	650	827	1039	1365	1710	2055	2410	2770	3125	3490
.58	504	650	827	1039	1375	1730	2090	2455	2825	3195	3570
.59	504	650	827	1039	1385	1755	2125	2505	2885	3270	3655
.60	504	650	827	1040	1400	1780	2160	2555	2945	3345	3745
.61	504	650	827	1045	1415	1805	2200	2605	3010	3420	3835
.62	504	650	827	1045	1430	1830	2240	2655	3075	3500	3925
.63	504	650	827	1050	1445	1860	2285	2710	3145	3580	4020
64	504	650	827	1060	1465	1895	2330	2770	3215	3665	4120
.65	504	650	827	1065	1490	1930	2375	2835	3295	3755	4225
.66	504	650	827	1075	1515	1965	2430	2900	3375	3850	4335
.67	504	650	827	1090	1540	2005	2485	2970	3465	3955	4455
.68	504	650	827	1105	1570	2050	2645	3050	3555	4065	4580
.69	504	650	827	1125	1600	2100	2615	3130	3655	4180	4710
.70	504	650	827	1150	1635	2155	2685	3220	3765	4305	4850
.71	504	650	827	1175	1680	2215	2760	3315	3875	4435	5000
.72	504	650	827	1205	1730	2280	2845	3420	3995	4575	5155
73	504	650	827	1245	1785	2355	2940	3530	4125	4720	5315
.74	504	650	827	1285	1845	2435	3040	3645	4260	4870	5490
.75	504	650	827	1330	1915	2520	3145	3770	4405	5035	5670
.76	504	650	827	1380	1985	2610	3255	3905	4560	5210	5865
.77	504	650	845	1440	2060	2710	3380	4050	4720	5390	6065
.78	504	650	885	1505	2150	2815	3510	4200	4895	5585	6275
79	504	650	935	1570	2240	2930	3650	4365	5080	5790	6500
.80	504	650	985	1645	2340	3060	3800	4535	5275	6010	6740
.81	504	650	1045	1730	2450	3195	3960	4720	5485	6240	7000
.82	504	650	1110	1820	2570	3345	4130	4920	5705	6490	7265
.83	504	650	1180	1925	2700	3505	4315	5130	5940	6750	7555
.84	504	650	1260	2035	2845	3670	4510	5355	6195	7025	7855
.85	504	650	1355	2160	3005	3855	4720	5595	6460	7320	8175
.86	504	685	1460	2300	3175	4055	4950	5845	6745	7630	8520
.87	504	765	1575	2455	3360	4270	5200	6120	7040	7960	8885
.88	504	855	1705	2620	3565	4510	5465	6415	7365	8310	9265
.89	504	960	1850	2800	3785	4765	5740	6720	7700	8680	9660
.90	504	1080	2015	3005	4020	5030	6045	7055	8060	9070	10070
.91	A 504	1215	2195	3230	4285	5330	6370	7405	8440	9470	10510
.92	530	1360	2395	3480	4560	5630	6700	7770	8840	9910	10990
.93	620	1525	2615	3750	4850	5950	7060	8160	9270	10370	11480
.94	725	1715	2860	4030	5160	6290	7430	8560	9710	10840	11980
.95	855	1935	3125	4325	5490	6650	7820	9000	10180	11350	12500
.96	1010	2175	3410	4640	5840	7030	8240	9460	10690	11880	13050
.97	1200	2445	3715	4975	6210	7440	8680	9950	11230	12440	13620
.98	1435	2745	4050	5330	6600	7880	9180	10470	11780	13000	14190
.99	1700	3065	4410	5720	7030	8340	9710	11040	12350	13570	14780
1.00	2010	3415	4790	6140	7500	8850	10270	11650	12960	14160	15370

TABLE 4 PRESSURE EXERTED BY CARBONIC ACID ON THE WALLS OF THE CONTAINING VESSEL, IN POUNDS PER SQUARE INCH[1]

Density, Water at 39.2°F=1	Temperature in Degrees Fahrenheit								
	32°	40°	50°	60°	70°	80°	90°	100°	110°
.50	504	565	650	744	849	965	1095	1245	1395
.51	504	565	650	744	849	965	1100	1250	1405
.52	504	565	650	744	849	965	1100	1255	1415
.53	504	565	650	744	849	965	1100	1260	1425
.54	504	565	650	744	849	965	1105	1265	1440
.55	504	565	650	744	849	965	1105	1270	1450
.56	504	565	650	744	849	965	1105	1280	1465
.57	504	565	650	744	849	965	1110	1290	1475
.58	504	565	650	744	849	965	1110	1300	1490
.59	504	565	650	744	849	965	1115	1310	1505
.60	504	565	650	744	849	965	1115	1320	1520
.61	504	565	650	744	849	965	1120	1330	1540
.62	504	565	650	744	849	965	1130	1340	1560
.63	504	565	650	744	849	965	1140	1355	1580
.64	504	565	650	744	849	965	1150	1375	1605
.65	504	565	650	744	849	965	1160	1395	1630
.66	504	565	650	744	849	965	1175	1415	1660
.67	504	565	650	744	849	965	1190	1440	1690
.68	504	565	650	744	849	975 B	1205	1465	1725
.69	504	565	650	744	849	990	1230	1495	1765
.70	504	565	650	744	849	1005	1255	1530	1810
.71	504	565	650	744	849	1025	1285	1570	1855
.72	504	565	650	744	849	1050	1320	1610	1910
.73	504	565	650	744	849	1080	1360	1660	1970
.74	504	565	650	744	849	1110	1405	1720	2035
.75	504	565	650	744	849	1145	1460	1785	2110
.76	504	565	650	744	870	1190	1515	1850	2190
.77	504	565	650	744	910	1240	1575	1925	2275
.78	504	565	650	744	955	1295	1645	2005	2370
.79	504	565	650	744	1005	1355	1715	2090	2470
.80	504	565	650	744	1060	1425	1795	2185	2575
.81	504	565	650	750	1120	1500	1885	2290	2695
.82	504	565	650	800	1190	1585	1985	2405	2825
.83	504	565	650	860	1260	1675	2095	2530	2965
.84	504	565	650	935	1345	1780	2215	2665	3115
.85	504	565	650	1020	1445	1895	2345	2815	3285
.86	504	565	685	1110	1550	2020	2490	2980	3465
.87	504	565	765	1210	1665	2155	2650	3160	3665
.88	504	565	855	1325	1800	2310	2825	3360	3880
.89	504	565	960	1450	1950	2485	3020	3565	4110
.90	504	655	1080	1595	2120	2675	3235	3800	4355
.91	A 504	765	1215	1755	2305	2880	3470	4050	4625
.92	530	880	1360	1925	2505	3115	3720	4320	4910
.93	620	1005	1525	2120	2740	3375	3995	4610	
.94	725	1145	1715	2340	2990	3645	4280	4920	
.95	855	1320	1935	2590	3255	3935	4585		
.96	1010	1515	2175	2860	3545	4240	4910		
.97	1200	1740	2445	3150	3850	4565			
.98	1435	2005	2745	3480	4195	4915			
.99	1700	2305	3065	3825	4555				
1.00	2010	2635	3415	4200	4940				

[1] The density here given is the combined density of the liquid and the overlying vapor for the portion of table lying above AB, and is the same as the proportion of filling in pounds of carbonic acid per pound of water capacity.

TABLE 4—Continued

Temperature in Degrees Fahrenheit.									
120°	130°	140°	150°	160°	170°	180°	190°	200°	212°
1550	1710	1870	2030	2185	2345	2505	2665	2825	3020
1565	1730	1895	2055	2220	2385	2550	2715	2880	3080
1580	1750	1915	2085	2255	2425	2595	2765	2930	3140
1600	1770	1940	2115	2290	2465	2640	2815	2990	3205
1615	1790	1970	2145	2325	2505	2685	2870	3050	3270
1630	1810	1995	2180	2365	2550	2735	2925	3110	3340
1650	1835	2025	2215	2405	2600	2790	2980	3175	3410
1670	1860	2055	2255	2450	2650	2845	3045	3245	3490
1690	1885	2090	2295	2495	2705	2905	3115	3320	3570
1710	1915	2125	2335	2545	2760	2970	3185	3395	3655
1735	1945	2160	2380	2595	2815	3035	3255	3475	3745
1760	1975	2200	2425	2645	2875	3100	3330	3555	3835
1785	2010	2240	2475	2700	2940	3170	3405	3640	3925
1815	2045	2285	2525	2760	3005	3240	3485	3725	4020
1845	2080	2330	2575	2820	3070	3315	3570	3815	4120
1875	2120	2375	2630	2885	3140	3395	3655	3910	4225
1910	2165	2430	2690	2950	3215	3480	3750	4010	4335
1950	2210	2485	2755	3025	3300	3570	3850	4120	4455
1995	2265	2545	2825	3105	3390	3670	3955	4235	4580
2040	2325	2615	2900	3190	3485	3775	4070	4355	4710
2095	2385	2685	2985	3280	3585	3885	4190	4485	4850
2150	2455	2760	3070	3375	3690	4000	4315	4620	5000
2215	2530	2845	3165	3480	3800	4125	4450	4765	
2290	2610	2940	3265	3595	3925	4260	4590	4915	
2365	2700	3040	3375	3715	4055	4400	4740		
2450	2790	3145	3495	3845	4195	4545	4895		
2535	2890	3255	3620	3980	4340	4700			
2635	3000	3380	3755	4120	4495	4865			
2740	3120	3510	3895	4275	4665				
2850	3245	3650	4050	4440	4840				
2975	3385	3800	4210	4615					
3110	3530	3960	4390	4800					
3255	3685	4130	4575	5000					
3410	3855	4315	4775						
3575	4040	4510	4985						
3760	4240	4720							
3960	4450	4950							
4175	4680								
4410	4935								
4655									
4915									

TABLE 5 PORTION OF THE CYLINDER VOLUME THAT IS OCCUPIED BY THE LIQUID FOR DIFFERENT COMBINED DENSITIES OF THE LIQUID AND SATURATED VAPOR OF CARBONIC ACID

Combined Density, That of Water=1	Temperature in Degrees Fahrenheit.										
	32°	40°	50°	60°	70°	75°	80°	82°	84°	86°	88°
50	493	.498	.508	.523	.543	.555	.575	.587	.601	.629	.807
.51	505	511	.522	.539	.561	.575	.599	.614	.631	.667	.891
52	.517	.524	.536	.554	.579	.596	.623	.640	.661	.705	.975
.53	529	.536	.550	.569	.597	.616	.647	.667	.691	.742	1.
.54	541	.549	.564	.585	.615	.636	.671	.693	.721	.780	1.
.55	554	.562	.578	.600	.633	.657	.695	720	751	.818	1.
.56	.566	.575	.591	.616	651	.677	719	.746	.781	.856	1.
.57	578	.588	.605	.631	.669	.697	743	.772	.811	.894	1.
.58	590	.601	.619	.647	.687	.717	767	.799	.841	.932	1.
59	.602	.613	.633	.662	.705	.738	.791	.825	.871	.970	1
.60	615	.626	.647	.677	723	.758	.815	.852	.901	1.	1.
.61	.627	639	.661	.693	.741	.778	.839	.878	.931	1.	1.
.62	639	.652	.675	708	.759	.799	.863	.905	.961	1	1.
.63	.651	.665	.688	724	.778	.819	.887	.931	.991	1.	1.
.64	.663	.677	.702	.739	.796	.839	.911	.958	1.	1.	1
65	676	.690	716	.755	.814	.860	.935	.984	1.	1	1
66	688	.703	730	.770	.832	.880	.959	1.	1	1	1.
67	700	716	.744	785	.850	.900	.983	1.	1.	1.	1.
.68	712	729	758	.801	.868	.921	1.	1.	1.	1	1.
69	724	741	.771	.816	.886	.941	1.	1.	1.	1	1.
70	737	754	785	.832	904	.961	1.	1.	1	1.	1.
71	749	767	799	.847	.922	.982	1.	1.	1.	1	1.
72	.761	.780	.813	.863	.940	1.	1.	1.	1.	1.	1.
73	773	793	.827	.878	.958	1.	1	1	1	1	1.
74	785	.805	.841	.894	.976	1.	1.	1.	1.	1	1.
.75	798	.818	855	.909	.995	1.	1.	1.	1.	1.	1
.76	.810	.831	.868	.924	1.	1.	1.	1.	1.	1.	1.
77	.822	.844	.882	.940	1.	1.	1.	1.	1.	1.	1.
.78	.834	.857	.896	.955	1.	1.	1.	1.	1.	1.	1.
79	.846	.869	.910	.971	1.	1.	1.	1.	1.	1.	1.
.80	.859	.882	.924	.986	1.	1.	1.	1.	1.	1.	1
.81	.871	.895	.938	1.	1.	1.	1.	1.	1.	1.	1.
.82	883	.908	.952	1.	1.	1.	1.	1.	1.	1.	1.
.83	.895	.921	.965	1.	1.	1.	1.	1.	1.	1.	1.
.84	.907	.933	.979	1.	1.	1.	1.	1.	1.	1.	1.
.85	.920	.946	.993	1.	1.	1.	1.	1.	1.	1.	1.
90	.980	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
95	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.
1.00	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.	1.

TABLE 5—Continued

Temperature in Degrees Centigrade.										
0°	5°	10°	15°	20°	22°	24°	26°	28°	30°	31°
.493	.499	.508	.521	.538	.546	.555	.569	.588	.629	.750
.505	.512	.522	.537	.556	.564	.576	.592	.615	.667	.819
.517	.525	.536	.552	.573	.583	.596	.615	.642	.705	.889
.529	.538	.550	.567	.591	.602	.616	.638	.668	.742	.958
.541	.551	.564	.582	.608	.621	.637	.661	.695	.780	1.
.554	.564	.578	.598	.625	.639	.657	.684	.722	.818	1
.566	577	.591	.613	.643	.658	.678	.706	.749	.856	1.
.578	.590	.605	.628	.660	.677	.698	.729	.776	.894	1.
.590	.603	.619	.643	.678	.695	.718	.752	.803	.932	1.
602	.615	.633	.659	.695	714	.739	.775	.830	.970	1.
.615	.628	.647	.674	713	.733	.759	.798	.857	1.	1.
627	.641	.661	.689	730	.751	780	.821	.884	1.	1.
.639	.654	.675	.704	.747	.770	.800	.844	.911	1.	1.
.651	.667	.688	.720	.765	.789	.820	.867	938	1.	1.
.663	.680	.702	.735	.782	.807	.841	.890	.965	1.	1.
.676	.693	.716	.750	.800	.826	.861	.913	992	1.	1.
.688	.706	730	.765	.817	.845	.882	.936	1.	1.	1.
.700	.719	.744	.780	.834	.864	.902	.959	1.	1.	1.
.712	.732	.758	.796	.852	.882	.922	.982	1.	1.	1.
.724	.745	.771	.811	.869	.901	943	1.	1.	1.	1.
.737	.757	.785	.826	.887	920	963	1	1.	1.	1.
749	.770	.799	.841	.904	.938	984	1.	1.	1.	1.
.761	.783	.813	.857	.922	.957	1.	1.	1.	1.	1.
.773	.796	.827	.872	.939	.976	1.	1.	1.	1.	1.
785	.809	.841	.887	.956	.994	1.	1.	1.	1.	1.
.798	.822	.855	.902	.974	1.	1.	1.	1.	1.	1.
.810	.835	.868	.918	.991	1.	1.	1.	1.	1.	1.
.822	.848	.882	.933	1.	1.	1.	1.	1.	1.	1.
.834	.861	.896	.948	1.	1.	1.	1.	1.	1.	1.
.846	.874	.910	.963	1.	1.	1.	1.	1.	1.	1.
.859	.886	.924	.979	1.	1.	1.	1.	1.	1.	1.
.871	.899	.938	.994	1.	1.	1.	1.	1.	1.	1.
.883	.912	.952	1.	1.	1.	1.	1.	1.	1.	1.
.895	.925	.965	1.	1.	1.	1.	1.	1.	1.	1.
.907	.938	.979	1.	1.	1.	1.	1.	1.	1.	1.
.920	.951	.993	1.	1.	1.	1.	1	1.	1.	1.
.980	1.	1.	1.	1.	1.	1	1.	1.	1.	1.
1.	1.	1.	1	1.	1.	1.	1.	1.	1.	1.
1.	1.	1.	1	1.	1.	1.	1.	1.	1.	1.

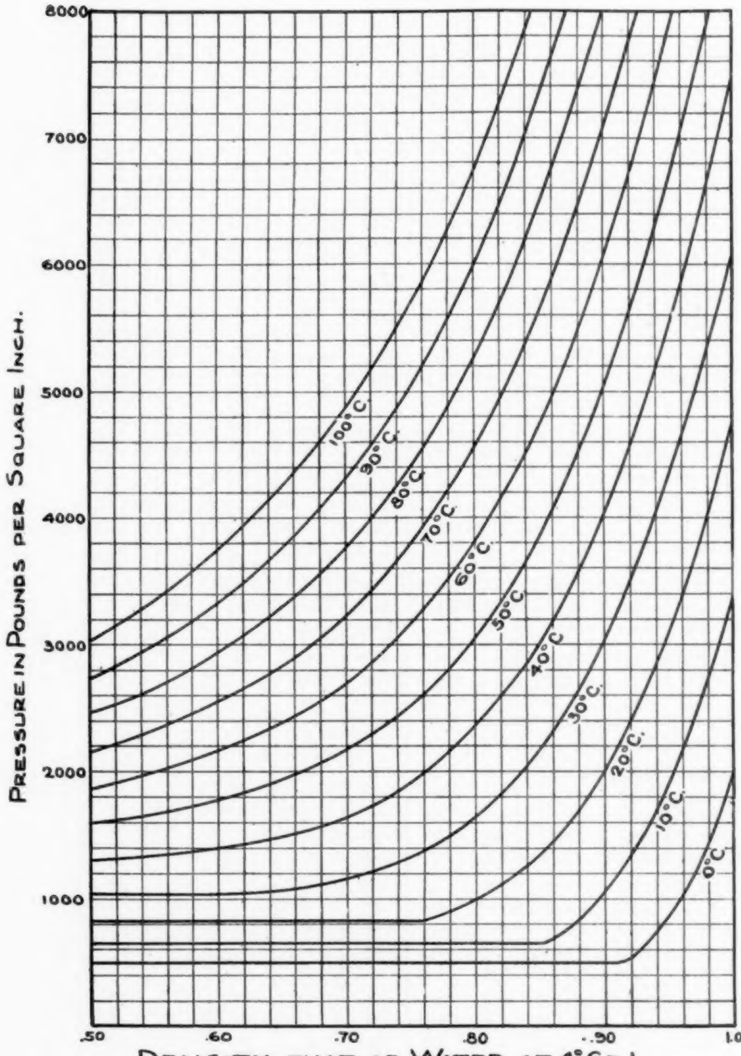

Fig. 2 Pressure Exerted by Carbonic Acid on the Walls of the Containing Vessel, in Pounds per Square Inch

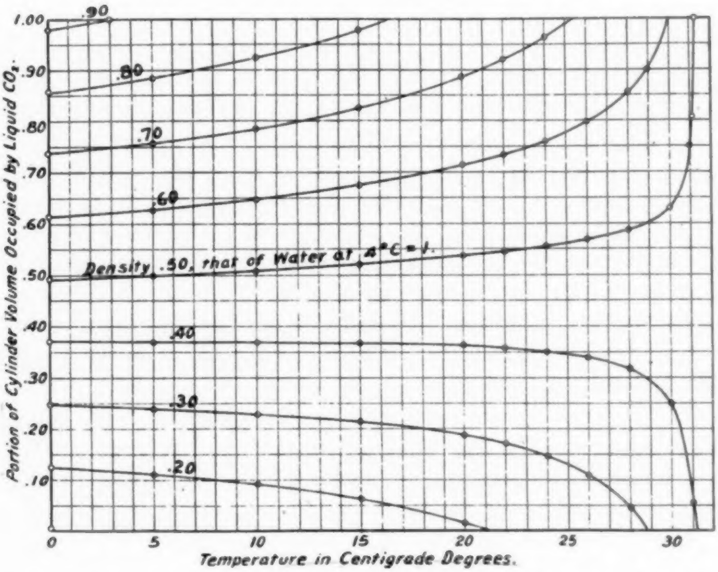

Fig. 3 Showing that for Combined Densities Greater than the Critical Density an Increase in Temperature is Attended by an Increase in the Liquid Volume of Carbonic Acid Contained in a Cylinder, While for Densities Less than the Critical the Law is Reversed, Increase in Temperature Being Attended by Decrease in Liquid Volume

PART 2 ECONOMIC STORAGE FOR TRANSPORTATION

Experiments on Carbonic Acid when under the Conditions of Commercial Storage for Transportation

73 All of the tables in Part 1 of this paper are based upon careful laboratory experiments conducted upon small quantities of purified and dried carbon dioxid. Those who may desire to use these scientific tables for either the design of new carbonic acid cylinders, or for the investigation of the safety of cylinders that are now in use, should have a clear understanding of what modifications, if any, should be made in these tables, when applying them to such purposes.

74 Commercial carbonic acid is generally saturated with water vapor, and will also be found to contain some air. Physicists apparently have not determined the pressures resulting from mixtures of gases and vapors under the conditions of the commercial storage of carbonic acid. The author, therefore, conducted experiments on commercial carbonic acid when under the precise conditions of storage for transportation. He has thus verified the scientific tables contained in Part 1, in so far as they bear directly upon the design of carbonic acid cylinders.

75 This experimental investigation was carried out in the mechanical laboratory of the Engineering School of the University of Pittsburg, by Frank P. Kramer, assisted by William M. Cooper, the whole being conducted under the immediate direction of the author.

GENERAL SCHEME OF EXPERIMENTS

76 The experiments were conducted according to the following general scheme: (1) a carbonic acid cylinder charged in the ordinary commercial way, was placed in a water jacket, which was provided with means for maintaining a constant known temperature; (2) samples of the carbonic acid were drawn off and analyzed for percentage of gaseous residue, or air content; (3) a pressure gage was then attached to the cylinder and the pressure determined corresponding to the known temperature; (4) the cylinder was then removed from the water jacket, dried externally, and weighed on an equal-arm balance; and (5) the cylinder was returned to the water jacket, where a certain definite quantity of acid was slowly discharged, and the

experiment was then repeated until the cylinder became fully discharged, the final weight obtained being that of the empty cylinder.

WATER JACKET

77 This was perhaps the most important piece of apparatus especially constructed for these experiments. It consisted as shown in Fig. 4, of a heavy seamless steel cup, which was protected from external radiation by means of a non-conducting covering. This piece of apparatus was designed with particular care with reference to keeping all parts of the carbonic acid cylinder under test at a constant predetermined temperature. Heat could be gradually imparted to the water jacket by means of the gas burner shown at the bottom, or heat could be gradually withdrawn by means of a cool water supply, not shown, at the top, the surplus water passing away automatically by means of an overflow. A very important feature was the longitudinal division of the water jacket into two parts, by which means the small propeller shown could produce a positive and rapid circulation of the water throughout the length of the jacket By these means no difficulty whatever was had in maintaining in all parts of the water jacket a predetermined temperature, within 0.1 deg. fahr.

78 The thickness of the wall of the seamless steel cup constituting this water jacket was made sufficient to protect the attendants from injury in case the carbonic acid cylinder under test should accidentally rupture.

SCHEME OF ANALYSIS FOR AIR CONTENT

79 The maturing of a satisfactory scheme for obtaining both readily and accurately the gaseous residue in the commercial carbonic acid, under the conditions of storage for transportation, was at first a source of considerable anxiety to the author. The following scheme, however, proved to be entirely satisfactory.

SELECTION OF A SAMPLE

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PART 2 ECONOMIC STORAGE FOR TRANSPORTATION

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be found to consist, at ordinary factory or laboratory temperatures, of a liquid portion overlaid by the vapor of carbonic acid. It is known of course in a general way that the air contained in the cylinder is unequally distributed within the liquid and the overlying vapor.

81 After due consideration of the matter it was decided that the best practical means for obtaining a representative sample of gas from a commercial cylinder would be to place the cylinder in a water jacket and elevate its temperature well above the critical temperature of the acid, thus converting the entire contents of the cylinder into a homogeneous gaseous mixture. This method for obtaining a representative sample of commercial carbonic acid has been thoroughly tested under the direction of the author, and has been found to be strictly reliable.

82 The truth of the above statement becomes apparent from an inspection of Table 6, which shows the results obtained by taking samples from the top of a commercial carbonic acid cylinder, at regular intervals while the cylinder was being slowly discharged, first, when the carbonic acid was above its critical temperature of 88.4 deg. fahr., and second, when below this temperature. This table shows that while the carbonic acid was being discharged at 110 deg. fahr. the successive analyses gave a constancy of gaseous residue within one-tenth of one per cent, and that while being discharged at 50 deg. fahr. the analyses gave a gaseous residue that varied from 8.2 to 0.2 per cent. That is to say, while the former gave constant results within the limit of accuracy of the apparatus used, the latter showed an extreme variation of about 4000 per cent.

SAMPLING CYLINDER FOR GAS ANALYSIS

83 While sampling commercial carbonic acid cylinders for gas analysis there should be kept in mind the following: First, that while below the critical temperature of 31.4 deg. cent., or 88.4 deg. fahr., the contents of a carbonic acid cylinder, as fully explained elsewhere in this paper, consists of a liquid portion overlaid by the vapor of carbonic acid, just as in a steam boiler the water is overlaid by steam; second, that the air is unequally distributed in the liquid carbonic acid and in its overlying vapor, thus rendering it impracticable to secure a representative sample for analysis from a cylinder when the contents are below the critical temperature of the carbonic

acid; third, that when above the critical temperature, if the cylinder has stood sufficiently long for complete diffusion, the contents of a commercial carbonic acid cylinder will consist of a homogeneous mixture of gaseous carbon dioxid with any impurities that are present in the gaseous or vaporous state, such as air and water vapor.

84 It is apparent then, from what has been given, especially from Table 6, that the only really reliable method for sampling a commercial carbonic acid cylinder is, first, to bring its temperature well above the critical temperature of carbon dioxid, or to say 100 deg. fahr., and then hold the cylinder at such temperature for a sufficient length of time for the complete diffusion of any gaseous or vaporous impurities that may be present before drawing off a sample for analysis.

85 Water in the liquid form and liquid lubricants coming over from the compressors, if these be present, will be found collected at the bottom of the cylinder and should be dealt with accordingly.

PRESSURE REGULATOR

86 The pressure regulator shown in Fig. 4 was devised to overcome the difficulty attendant upon supplying gas, when stored under pressures varying from 1000 to 2000 or more lb. per square inch, to a burette, where the pressure is atmospheric. It is clearly impracticable to supply the gas directly from the cylinder valve to the burette without the use of some sort of pressure regulator, because no nice adjustment of flow can be effected by means of the regular cylinder valve, because of the violent spitting action due to "freezing" in the valve.

87 The regulator used, as shown in the figure, consisted of placing a tee in the tube leading from the cylinder valve to the burette, the side branch of the tee terminating in an open glass tube hanging vertically in a bath of water. By this simple device the gas supply to the burette was kept at a constant pressure, all variable excess of gas escaping automatically from the lower end of the vertical glass tube. This exceedingly simple regulator made it possible to fill the burette with the greatest ease and prevented any possibility of accident to the burette while being filled.

BURETTE FOR THE GAS ANALYSIS

88 The burette for the gas analysis consisted of a modified form of Bunte's apparatus, adapted to making rapid determinations of gaseous residues to the nearest 0.1 per cent. The regulator and burette were arranged and manipulated so that the sample of carbonic acid for analysis nowhere bubbled up through water, but was made to enter the burette from the top by means of water displacement, the absorption being effected in the usual way by means of potassium hydroxid. This apparatus gave results that were uniformly consistent within one-tenth of one per cent of gaseous residue.

EFFECTS OF GASEOUS IMPURITIES UPON THE PRESSURES EXERTED

89 These are shown very clearly in Fig. 6. This chart is for a temperature of 120 deg. fahr., and shows the different pressures corresponding to combined densities ranging from 0.5 to 0.7. Curve A is plotted from the values given in Table 4, and shows the relation of pressure to temperature and density for pure carbon dioxid. Curve B is plotted from experiments on commercial carbonic acid contained in commercial cylinders, when the carbonic acid is saturated with water vapor and the gaseous residues or air content is 0.7 per cent. Curve C, ditto, when the gaseous residue or air content is 1.7 per cent.

90 For the most economic conditions of commercial storage and transportation, then, 1 per cent of air content corresponds to an increase in the fluid pressure of about 60 lb. per square inch.

PRESSURE GAGE

91 The gage used in connection with these experiments was a Shaw differential-piston mercury-gage, having a capacity of 3000 lb. per square inch. The scale of this gage was calibrated by means of a dead-weight tester, having a capacity of 1000 lb., the scale being pointed at 500, 750 and 1000 lb., and these in turn duplicated up to 3000 lb. All pressures recorded in this paper as obtained from carbonic acid cylinders were corrected in accordance with this calibration.

92 The dead-weight tester used in calibrating the pressure gage was examined by calipering the plunger and by weighing the individual weights used and was found to be substantially correct.

WEIGHING CYLINDERS AND ACID

93 The commercial carbonic acid experimented with was contained in steel cylinders that were nominally 5 in. inside diameter and 50½ in. long. These cylinders were made expressly for these experiments of ductile steel. The cylinders were seamless and were constructed of sufficient thickness of wall to secure about double the safety of the ordinary commercial cylinder of the same capacity. This was deemed necessary as a proper safeguard in view of the fact that these cylinders were to be subjected to the highest pressure conditions reached by commercial charged cylinders. In order, therefore, to obtain correct weights of the acid experimented with it was necessary to make accurate gross weighings in the neighborhood of from 150 to 175 lb.

94 It was thought at first that the weighings could be made on a platform scale, built by a well-known scale firm for laboratory purposes, having a capacity of 300 lb. and capable of being read to 0.01 lb. In an attempt to calibrate this platform scale, before being put into use, it was immediately discovered that it was incapable of giving accurate readings. This became apparent when a standard ten pound weight, when placed upon its platform, gave readings which varied by as much as 43 times the smallest reading of the beam, which variation appeared to be caused by a change in the leverage ratio which was due to a shifting of the parts underneath the platform.

95 After due consideration of the matter it was decided to weigh the charged and empty carbonic acid cylinders in the manner illustrated in Fig. 5, the inherent inaccuracies of the platform scale rendering its use inadmissible for scientific work where, as in this case, the gross weights are many times as great as the corresponding net weights to be determined. As shown in the figure the weighings were made on an equal-arm balance, especially constructed of sufficient capacity for the purpose, and provided with suitable means for suspending the carbonic acid cylinder from the knife edge of one end, while the usual form of scale pan was suspended from the knife edge of the other end. All weighings were recorded to the nearest .01 lb. only, as this was deemed quite exact enough for the present investigation, although weighings could of course have been made with much greater precision.

WATER CAPACITY OF CYLINDER

96 The water capacities of the cylinders used were obtained as follows: first, the cylinder, while being suspended in a vertical position, was filled with distilled water, after which heat was applied until the water boiled, thus expelling the air in solution; second, the cylinder was then placed in the water jacket shown in Fig. 4 and cooled to a temperature approximating that of the laboratory hydrant water; third, distilled water, separately boiled and cooled in a flask, was poured into the cylinder to make up the deficiency due to shrinkage of water volume while the cylinder was being cooled; fourth, the cylinder was then removed from the water jacket, surface dried, and weighed in the manner shown in Fig. 5; and, fifth, the cylinder was then emptied, dried internally, reweighed, and the water capacity obtained by taking the difference of cylinder weights thus obtained; the final reduction to standard conditions being made by means of Castell Evans Physico-Chemical Tables, 1902 edition.

97 An investigation was made to discover what error in water capacity would result from the use of ordinary Pittsburg hydrant water without any corrections whatever being made for dissolved air and other impurities, reduction to vacuo and maximum density, etc. The results of this investigation showed that for commercial purposes it is not necessary to observe these refinements, when obtaining the water capacity of a cylinder, since the errors resulting from the use, without any corrections whatever, of ordinary hydrant water will not exceed those due to the ordinary inaccuracies of commercial operations.

98 For commercial cylinders, then, the proper weight of carbonic acid to be charged, as based upon the tables contained in this paper, may be had as follows: first, obtain the water capacity of the cylinder by subtracting its weight when empty from its weight when filled with ordinary clear stream or hydrant water; and, second, multiply the resulting water capacity in pounds, by the mean density of the carbonic acid, Table 7, corresponding to the greatest temperature to which the charged cylinder is to be subjected.

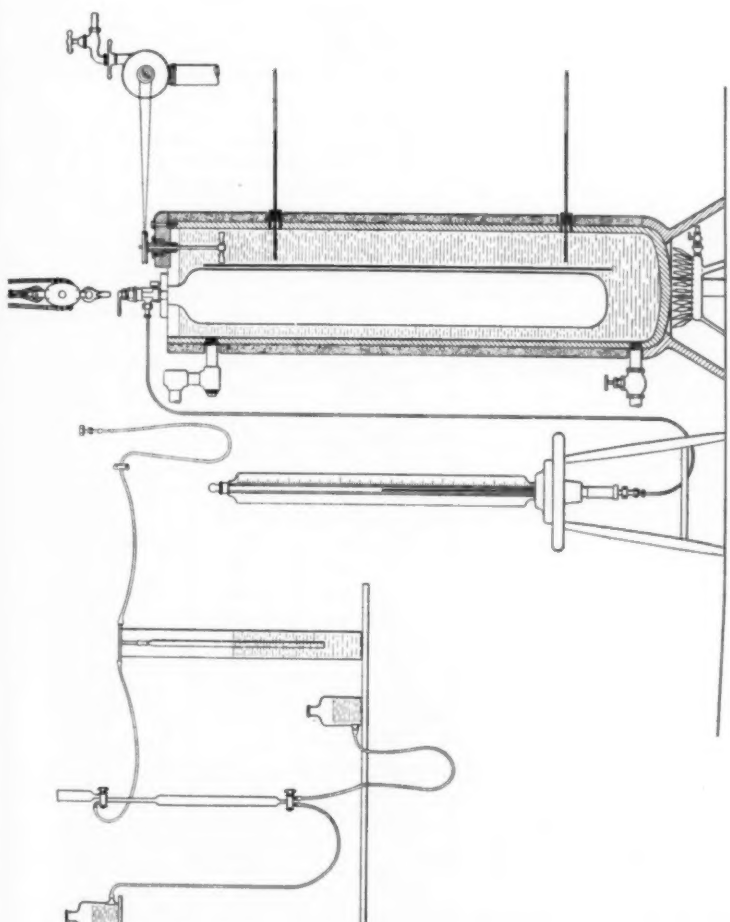


Fig. 4 Apparatus Used in Experiments on Carbonic Acid Under the Conditions of Commercial Storage for Transportation

TABLE 6 SHOWING CONSTANCY OF AIR CONTENT OF COMMERCIAL CARBONIC ACID WHEN SAMPLES FOR ANALYSIS ARE DRAWN OFF AT 110 DEG. FAHR. COMPARED WITH THE VARIABLE AIR CONTENT OBTAINED FROM SAMPLES DRAWN OFF AT 50 DEG. FAHR., THE FORMER ABOVE AND THE LATTER BELOW THE CRITICAL TEMPERATURE OF 88.4 DEG. FAHR.[1]

First Filling of Cylinder "B" Slowly Discharged at Temperature of 110 °F.			Second Filling of Cylinder "B" Slowly Discharged at Temperature of 50 °F.		
Weight of CO_2 Pounds	Density of CO_2 Water = 1.	Air in CO_2 Per cent.	Weight of CO_2 Pounds.	Density of CO_2 Water = 1.	Air in CO_2 Per cent.
22.97	.690	.75	22¼	.67	8.2
19.92	.599	.75	21½	.65	6.8
19.64	.590	.8	20½	.62	5.8
18.97	.570	.8	19	.57	4.5
17.98	.540	8	18¾	.55	3.3
17.10	.514	.8	17¼	.52	2.5
15.90	.478	.75	16	.48	1.9
14.64	.440	.8	15	.45	1.5
13.28	.399	.8	14	.42	1.1
12.02	.361	.8	12½	.38	.8
10.64	.320	.8	11½	.35	.6
9.28	.279	.75	10½	.32	.5
8.07	243	.8	9¾	.28	.4
6.38	192	.8	7¼	.22	.3
5.45	164	75	4¾	.14	.2
4.49	135	.8			
3.67	.110	.8			
2.34	.071	.8			
1.29	.039	.8			
0.00	.000	.75			

[1] No determination was made for the actual air content for the second filling of cylinder B. When making comparison it should not therefore be assumed to be the same as for the first filling.

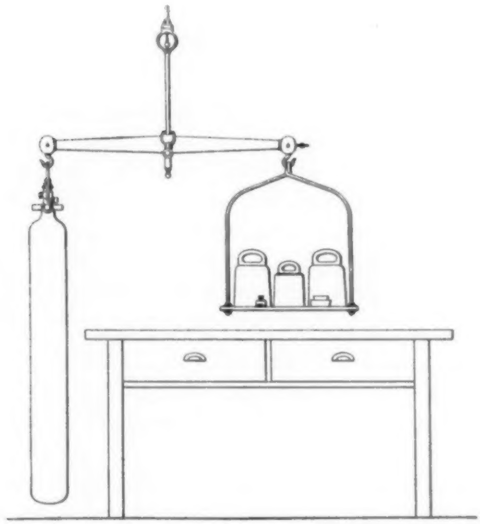


FIG. 5 METHOD OF WEIGHING THE CARBONIC ACID CYLINDERS

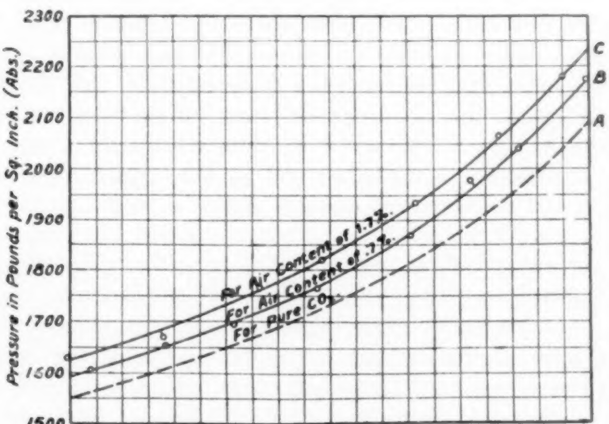


FIG. 6 SHOWING FOR A TEMPERATURE OF 120 DEG. FAHR. THE DIFFERENT PRESSURES CORRESPONDING TO COMBINED DENSITIES RANGING FROM 0.50 TO 0.7

CURVE A IS PLOTTED FROM THE VALUES GIVEN IN TABLE 4, AND SHOWS THE RELATION OF PRESSURE TO TEMPERATURE AND DENSITY FOR PURE CARBON DIOXID. CURVE B IS PLOTTED FROM EXPERIMENTS ON COMMERCIAL CARBONIC ACID CONTAINED IN COMMERCIAL CYLINDERS, WHEN THE GASEOUS RESIDUE OR AIR CONTENT IS 0.7 PER CENT. CURVE C, DITTO, WHEN THE GASEOUS RESIDUE OR AIR CONTENT IS 1.7 PER CENT. FOR THE MOST ECONOMIC CONDITIONS OF STORAGE, THEN, 1 PER CENT OF AIR CONTENT CORRESPONDS TO AN INCREASE IN THE FLUID PRESSURE OF 60 LB. PER SQUARE INCH

PART 3 DESIGN OF CARBONIC ACID CYLINDERS

99 The foregoing tables of the physical properties of carbonic acid are of sufficient scope to permit of a full inquiry into the economics of carbonic acid storage and transportation. It was chiefly for the purpose of providing sufficient data for obtaining the important conditions of minimum cylinder weight that the preparation of these tables was undertaken. These conditions will be considered, first, when the effects of the thickness of the cylinder wall upon the storage capacity are neglected, and, second, when these effects are taken into account.

MINIMUM WEIGHT OF CYLINDER, NEGLECTING EFFECTS OF THICKNESS

100 Neglecting the effects of the thickness of the cylinder wall upon the storage capacity, which for the actual conditions of storage for transportation will necessarily be small, it is evident that the minimum weight of the cylinder per unit weight of the carbonic acid contained will correspond to the minimum quotient resulting from dividing the pressure of the carbonic acid by its corresponding density. This becomes evident when it is considered, first, that the weight of the cylinder, for the assumed conditions, will be directly as the pressure exerted by the carbonic acid, since the weight of the cylinder is proportional to the thickness, and the thickness in turn is proportional to the pressure; and, second, that the weight of the cylinder per unit weight of carbonic acid contained will be inversely as the density of the contained acid, that is to say, other things being equal, the weight of a cylinder will become less relatively as the density of the contained acid becomes greater. Now combining these two fundamental relations we are led to the conclusion that the weight of a cylinder relative to that of its contained carbonic acid is a minimum when the pressure of the acid divided by its corresponding density is a minimum.

101 It should be noted here that the above relations apply directly to the shell or cylindrical portion of the carbonic acid cylinder; and also to the whole cylinder, including the heads, when these have circular axial sections that are of the same theoretical strength as the shell.

MINIMUM VALUES OF PRESSURES DIVIDED BY CORRESPONDING DENSITIES

102 Fig. 8 was prepared from Table 3 by dividing the different tabular pressures by their corresponding densities; after which the resulting quotients were plotted, as shown, to a vertical scale of pressure divided by density $\frac{P}{G}$, and to a horizontal scale of pressure in pounds per square inch.

103 A glance at this figure shows that, corresponding to each temperature, there is a well defined minimum quotient of pressure divided by density. For example, the minimum quotients for temperatures of 30, 40, 50, and 60 deg. cent. are seen to correspond to pressures of 1090, 1480, 1830, and 2140 lb. per square inch respectively; or to densities of 0.67, 0.65, 0.62, and 0.60 respectively.

104 Neglecting the effects of the thickness of the wall upon the storage capacity of the cylinder, which, as stated above will necessarily be small for actual commercial conditions, it is apparent that this chart, Fig. 8, shows approximately the conditions of minimum weight of cylinder per unit weight of carbonic acid contained.

105 The following is a more exact solution of this problem, in which the effects of the thickness of cylinder wall upon its storage capacity and the working fiber strength of the steel are taken into consideration.

FORMULA FOR WEIGHT OF STEEL IN SHELL PER POUND OF CARBONIC ACID CONTAINED

106 Let Fig. 7 represent a transverse section of a cylinder that contains carbonic acid under known conditions of density and pressure, the working fiber stress of the material also being known. It is desired to derive a formula that will show the relation of the weight of the shell of the cylinder to the weight of the carbonic acid contained. Let

D = outside diameter of the cylinder in inches.
W = weight of shell in pounds per lineal foot.
w = weight of carbonic acid contained in pounds per lineal foot of cylinder.
t = thickness of cylinder wall in inches.
V = volume in cubic inches of carbonic acid contained per lineal foot of cylinder.

G = specific gravity, or density of the carbonic acid, that of pure water at 4 deg. cent. = 1.
f = fiber stress in wall of cylinder, pounds per square inch.
p = internal fluid pressure, pounds per square inch.

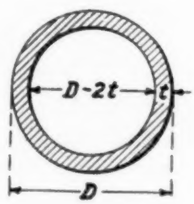


FIG. 7

Then the weight of the shell, or cylindrical portion of the carbonic acid cylinder, in pounds per lineal foot, will equal the continued product of the mean circumference $\pi\ (D-t)$, the thickness t, the length, 12, and the weight of a cubic inch of steel, 0.2833, or

$$W = 10.680\ (D-t)\ t$$

The internal volume of the cylinder in cubic inches per lineal foot or its capacity for carbonic acid, will be

$$V = \frac{\pi}{4}\ (D-2t)^2 \times 12 = 9.425\ (D-2t)^2$$

The weight of carbonic acid contained, in pounds per lineal foot of cylinder will equal the water capacity of the cylinder per lineal foot multiplied by the density of the carbonic acid, or

$$w = \frac{62.43}{1728}\ VG = 0.03613\ VG = 0.3405\ (D-2t)^2\, G$$

The number of pounds of steel in the cylinder, exclusive of the heads, for each pound of carbonic acid contained will then be

$$\frac{W}{w} = \frac{10.680\ (D-t)\ t}{0.3405\ (D-2t)^2\, G} = 31.37 \frac{(D-t)\ t}{(D-2t)^2\, G} \qquad (3)$$

107 In order to arrive at a formula that shall express this relation in terms of the internal fluid pressure and the resulting fiber stress in the cylinder wall we shall first divide both the numerator and denom-

inator of the last member of equation 3 by $(D-t)\,t$. Doing this we get

$$\frac{W}{w}=\frac{31.37}{\left(\frac{D}{t}+\frac{t}{D}-3\right)G} \tag{4}$$

which is a close approximate value of $\frac{W}{w}$, expressed in terms of the density of the carbonic acid and the relation of the thickness to the outside diameter of the cylinder, $\frac{t}{D}$ and $\frac{D}{t}$.

108 An investigation of the relative merits of the different formulae that have been published for the strength of tubes which are subjected to internal fluid pressures, has led the author to the belief that Clavarino's formula is the most reliable, when the tube wall is subjected to both the transverse and longitudinal stresses due to the internal fluid pressure; as is the case in a carbonic acid cylinder. Unfortunately this formula is too complex in form for direct substitution of values for $\frac{t}{D}$ and $\frac{D}{t}$ in equation 4.

109 The author, however, has produced an exceedingly close approximation to Clavarino's formula for values of $\frac{t}{D}$ less than 0.1; which fully covers the range in values of $\frac{t}{D}$ for the conditions of minimum weight of cylinder per unit weight of contained carbonic acid. This very close approximate formula, for steel carbonic acid cylinder conditions, gives the relations

$$\frac{D}{t}=2.325\,\frac{f}{p},\ \text{or}\ \ \frac{t}{D}=0.43\,\frac{p}{f} \tag{5}$$

Now substituting these values of $\frac{D}{t}$ and $\frac{t}{D}$ in equation 4 we get

$$\frac{W}{w}=\frac{31.37}{\left(2.325\,\frac{f}{p}+0.43\,\frac{p}{f}-3\right)G} \tag{6}$$

which expresses the relation of the weight of steel in one unit length of the shell to the weight of the carbonic acid contained in a unit length; this relation being expressed in terms of the density and pressure of the carbonic acid, and of the working fiber stress of the steel.

MOST ECONOMIC CONDITIONS OF STORAGE AND TRANSPORTATION

110 Formula 6 makes it possible to investigate fully the conditions that correspond to the minimum weight in the shell of a carbonic acid cylinder per unit weight of contained acid. The results of such an investigation are shown in Fig. 9, and in Table 7. Values for the density G, and the corresponding pressure p, for temperatures ranging from 90 to 140 deg. fahr., as taken from Table 4, were substituted in formula 6, together with 15 000 lb. per square inch for the working fiber stress in the cylinder wall. The results obtained are shown plotted in Fig. 9, to a vertical scale of weight of shell divided by weight of the acid contained, each per lineal foot, and to a horizontal scale of fluid pressure exerted by the carbonic acid, in pounds per square inch.

111 It will be observed that each temperature curve of this figure has a well defined minimum ordinate, or ratio of weight of shell to weight of contained acid. For example, when the carbonic acid cylinder and its contents are at a temperature of 110 deg. fahr. this minimum ordinate corresponds to a density, or proportion of filling by weight in water capacity G, of 0.628; and to a fluid pressure within the cylinder p, of 1575 lb. per square inch. These then are the most economic conditions of storage and transportation of carbonic acid, when the maximum temperature of the cylinder while charged is 110 deg. fahr.; for it is apparent from the form of the curve that, for any given degree of safety, either a less or a greater proportion of filling will require a greater weight of steel in the cylinder wall per unit weight of contained carbonic acid than will be required for the most economic conditions.

112 Similarly the most economical conditions of storage for transportation have been obtained for fiber stresses ranging from 15 000 to 30 000 lb. per square inch and for maximum storage temperatures ranging from 100 to 155 deg. fahr. Table 7 shows these results put into the most convenient form for the design, on a rational basis, of carbonic acid cylinders.

THICKNESS FACTORS FOR CARBONIC ACID CYLINDERS

113 Table 7 is based upon the results of laboratory experiments on purified and dried carbon dioxid. It is therefore directly applicable to the storage in steel cylinders of chemically pure carbonic acid.

114 In order to determine the applicability of Table 7 to the design of steel cylinders for the storage and transportation of commercial carbonic acid the author conducted a series of experiments on commercial carbonic acid under the precise conditions of commercial storage for transportation, an account of which will be found recorded in Part 2 of this paper. The results of these experiments show for commercial acid, fully saturated with water vapor, that an air content of one per cent will cause an increase in the thickness factors contained in Table 7 of about 4 per cent.

APPLICATION OF TABLE 7 TO THE DESIGN OF CARBONIC ACID CYLINDERS

115 It will be observed that the main entries of this table are thickness factors, each of which represents, for the condition of minimum weight of steel in the shell, the quotient obtained by dividing the thickness of the shell by its outside diameter, or $\frac{t}{D}$, both being expressed in inches.

116 Before attempting to apply this table to the design of commercial carbonic acid cylinders, it will be found necessary, of course, to determine, first, the greatest safe working fiber stress of the steel constituting the finished cylinder, and, second, the greatest temperature that the contents of the charged cylinder will reach under the ordinary conditions of the commercial storage and transportation of the acid.

117 Any discussion of these matters would be clearly outside the province of the present paper, which is distinctly a monograph on those physical properties of carbonic acid that have a bearing upon rational methods of cylinder design and, as well, upon investigations into the stresses to which charged cylinders are subjected.

118 Assuming, for instance, the maximum working fiber stress in the wall of a cylinder to be 17 000 lb. per square inch, and the maximum storage temperature of the carbonic acid to be 120 deg. fahr., we find from Table 7 the corresponding thickness factor, $\frac{t}{D}$, to be 0.0443. For these conditions, then, a cylinder of 5½ in. outside diameter should have a thickness t, equal 0.0443 by 5½, or 0.244 in.; which is approximately ¼ in.

CHARGING FACTORS FOR CARBONIC ACID CYLINDERS

119 At the bottom of Table 7 will be found a set of charging factors, for determining the quantity of acid to be charged into cylinders, that have been designed by use of the corresponding thickness factors. It will be observed that for any given maximum storage temperature the corresponding charging factor is expressed in three different ways, namely, first, as pounds carbonic acid per pound water capacity, which for the above example, where the maximum storage temperature is assumed to be 120 deg. fahr. would be 0.61; second, as pounds carbonic acid per cubic inch of cylinder capacity, which for the conditions of the above example would be 0.0220; and, third, as pounds carbonic acid per U. S. gallon of cylinder capacity, or for the above example, 5.07.

120 In order then to have a cylinder carry 20 lb. of acid at a maximum storage temperature of 120 deg. fahr., when the cylinder is designed according to Table 7, it should have a water capacity of $20 \div 0.61 = 32.8$ lb.; or it should have a cubic capacity of $20 \div 0.0220 = 910$ cu. in.; or, it should have a capacity of $20 \div 5.07 = 3.95$ U. S. gal.

APPLICATION OF TABLE 4 TO DETERMINING STRESSES IN CARBONIC ACID CYLINDERS

121 In Part 1 of this paper will be found a full explanation of Table 4, the entries of which are the absolute fluid pressures of pure carbonic acid, in pounds per square inch, corresponding to the different temperatures and densities tabulated, the densities being referred to pure water at its maximum density, all weighings being reduced to vacuo.

122 When applying this scientific table to ordinary commercial conditions, for weights of acid that are known to be accurate only to within a pound and to temperatures of the acid that are known to be accurate only to within 5 deg. fahr.; no correction need be applied to the values given in Table 4; unless the air content of the acid is excessive, in which case a correction should be made which, for temperatures ranging from 100 to 140 deg. fahr., will average about 60 lb. increase in tabular pressure for each per cent of air contained in the carbonic acid.

123 For calculating the fiber stress in the wall of a carbonic acid cylinder use formula 5 contained in Part 2 of this paper, which is

$$f = 0.43 \frac{pD}{t} \text{ where}$$

f = fiber stress in cylinder wall in pounds per square inch of material.
p = internal fluid pressure in pounds per square inch.
D = outside diameter in inches.
t = thickness of cylinder wall in inches.

124 It should be remarked here that this simple formula gives results which, for carbonic acid cylinder conditions, approximate exceedingly close to those obtained by use of Clavarino's theoretically correct formula which is

$$f = \frac{4}{3} \frac{D^2 + (D - 2t)^2}{[D^2 - (D - 2t)^2]} p \qquad (7)$$

125 A cylinder having a water capacity of 33 pounds when charged with 20 pounds of commercial acid, corresponding to a tabular density of $\frac{20}{33}$, or 0.61, and having an air content of 2 per cent, when at a temperature of 120 deg. fahr., will be subjected to a fluid pressure, according to Table 4, of 1760 + (60 × 2), or 1880 pounds per square inch. If now this cylinder have an outside diameter of $5\frac{1}{2}$ in. and a thickness of wall of $\frac{1}{4}$ in., then the fiber stress in the wall will be

$$f = 0.43 \frac{pD}{t} = 0.43 \frac{1880 \times 5.5}{.25} = 17\,800 \text{ lb. per sq. in.}$$

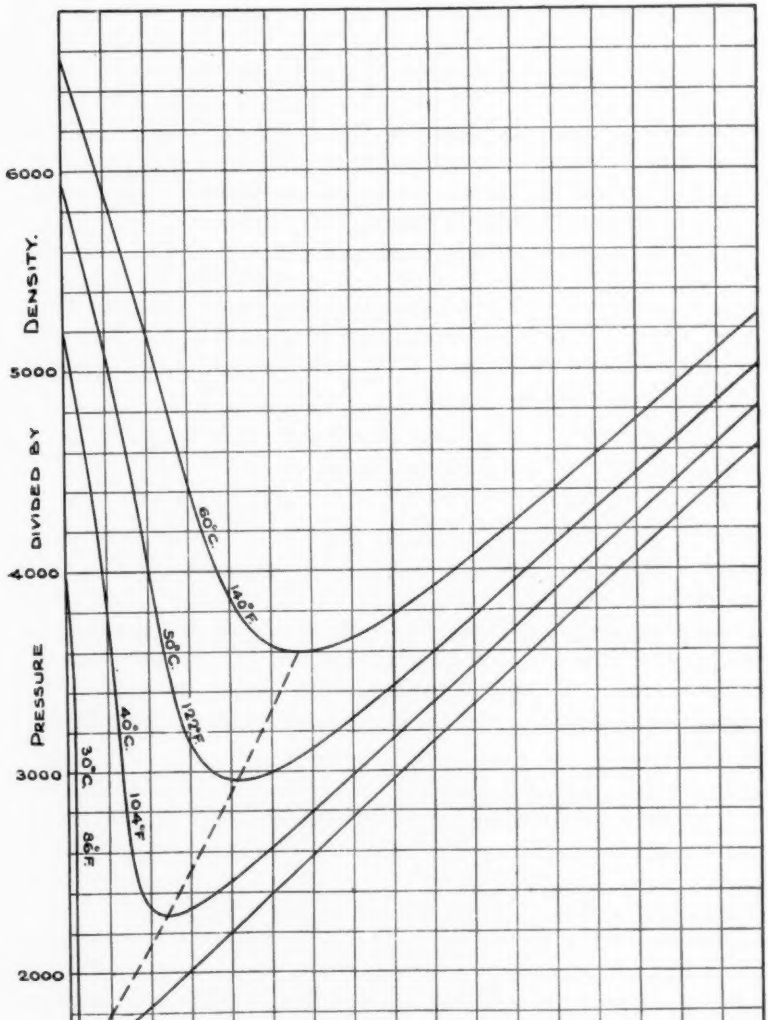


FIG. 8 CURVES SHOWING THE RELATION OF PRESSURE DIVIDED BY DENSITY TO THE PRESSURE OF CARBONIC ACID

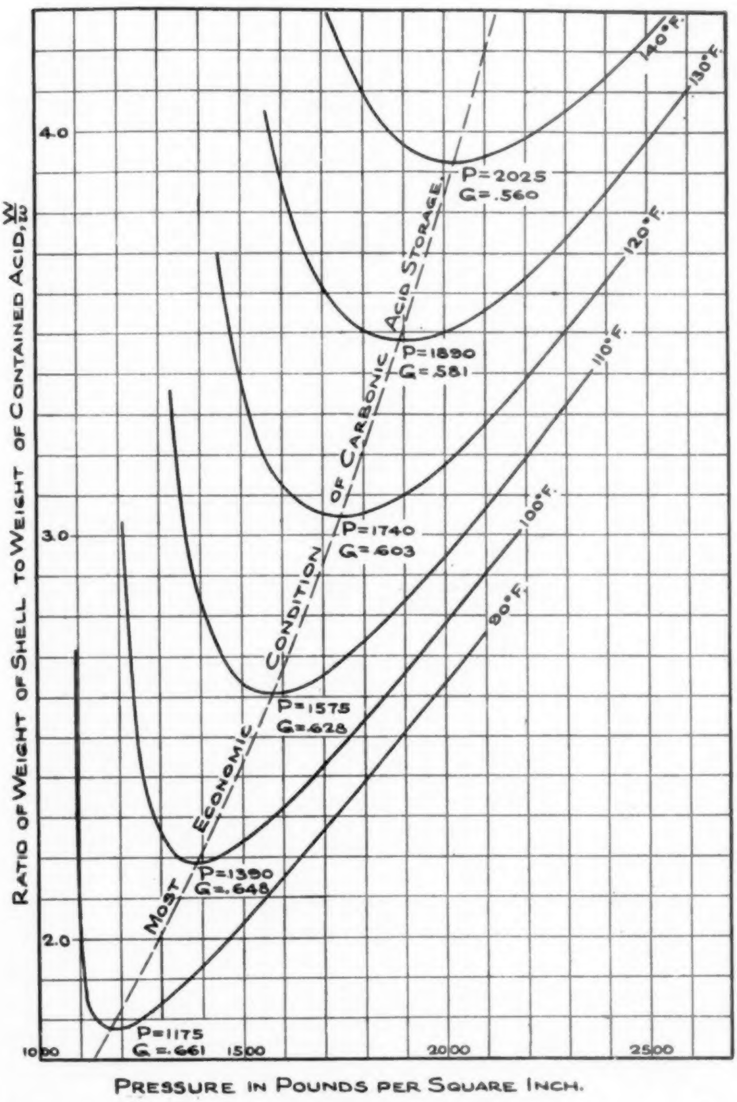


FIG. 9 MOST ECONOMIC CONDITION OF CARBONIC ACID STORAGE FOR FIBER STRESS OF 15 000 LB.

TABLE 7 THICKNESS AND CHARGING FACTORS FOR CARBONIC ACID CYLINDERS CORRESPONDING TO THE CONDITION OF MINIMUM WEIGHT OF STEEL IN THE SHELL PER UNIT WEIGHT OF ACID CONTAINED

RULE:—TO FIND THE THICKNESS OF CYLINDER WALL, MULTIPLY THE OUTSIDE DIAMETER BY THE THICKNESS FACTOR CORRESPONDING TO THE WORKING FIBER STRESS AND GREATEST STORAGE TEMPERATURE

Fiber Stress in Wall, Pounds per Sq. Inch.	GREATEST TEMPERATURE TO WHICH CHARGED CYLINDER IS SUBJECTED.											
	100°F	105°F	110°F	115°F	120°F	125°F	130°F	135°F	140°F	145°F	150°F	155°F
	FACTORS FOR THICKNESS OF CYLINDER WALL, OR t/D.											
15000	.0398	.0426	.0452	.0476	.0499	.0521	.0542	.0562	.0581	.0599	.0616	.0633
16000	.0375	.0401	.0425	.0449	.0470	.0491	.0511	.0530	.0547	.0564	.0581	.0597
17000	.0353	.0378	.0401	.0423	.0443	.0463	.0482	.0500	.0515	.0532	.0548	.0563
18000	.0334	.0357	.0379	.0400	.0419	.0438	.0455	.0472	.0487	.0503	.0518	.0533
19000	.0316	.0338	.0359	.0378	.0397	.0415	.0432	.0448	.0462	.0477	.0492	.0506
20000	.0300	.0321	.0341	.0359	.0377	.0394	.0410	.0425	.0439	.0454	.0468	.0481
21000	.0286	.0306	.0325	.0342	.0359	.0375	.0391	.0406	.0419	.0433	.0446	.0459
22000	.0273	.0292	.0310	.0327	.0344	.0359	.0374	.0388	.0401	.0414	.0427	.0439
23000	.0261	.0279	.0297	.0313	.0329	.0344	.0358	.0372	.0385	.0397	.0409	.0421
24000	.0251	.0268	.0285	.0300	.0316	.0330	.0344	.0357	.0370	.0381	.0393	.0405
25000	.0241	.0258	.0274	.0289	.0303	.0317	.0331	.0344	.0356	.0367	.0379	.0390
26000	.0232	.0248	.0264	.0278	.0292	.0306	.0319	.0331	.0343	.0354	.0365	.0376
27000	.0223	.0239	.0254	.0268	.0282	.0295	.0307	.0320	.0331	.0342	.0353	.0363
28000	.0215	.0231	.0245	.0259	.0272	.0285	.0297	.0309	.0320	.0331	.0341	.0351
29000	.0208	.0223	.0237	.0250	.0263	.0275	.0287	.0299	.0309	.0320	.0330	.0340
30000	.0202	.0216	.0229	.0242	.0254	.0266	.0278	.0289	.0299	.0310	.0320	.0329
These charging factors are substantially correct for fiber stresses from 15,000 to 30,000 lbs. per square inch	MEAN DENSITY, OR POUNDS OF CARBONIC ACID PER POUND WATER CAPACITY.											
	.65	.64	.63	.62	.61	.60	.59	.58	.57	.56	.55	.54
	MEAN WEIGHT OF CARBONIC ACID IN POUNDS PER CUBIC INCH.											
	.0235	.0232	.0228	.0224	.0220	.0216	.0212	.0208	.0205	.0201	.0198	.0196
	MEAN WEIGHT OF CARBONIC ACID IN POUNDS PER U.S. GALLON.											
	5.42	5.36	5.27	5.17	5.07	4.98	4.90	4.82	4.73	4.65	4.57	4.51

ADDENDUM

126 The present paper, it is believed, furnishes data on the physical properties of carbonic acid of sufficient scope for the design of carbonic acid cylinders on a rational basis and for the investigation of the safety of cylinders that are now in use. While complete in itself, this paper is based, so far as cylinder design and investigation for safety are concerned, upon assumed safe working fiber stresses of the material constituting the cylinder wall. The engineer who undertakes to design cylinders for the transportation of carbonic acid, in addition to what is here given, should also be in possession of sufficient knowledge of the properties of the different steels in their relation to processes of cylinder manufacture, heat treatment, and usage to which the commercial cylinders are subjected, to be able to select the material best suited to cylinder construction.

127 There appears to be the greatest diversity of opinion regarding the most suitable material for cylinder construction. For example, in Great Britain, those engaged in compressing carbonic acid for transportation are following substantially the recommendations of the report to Parliament in 1896 of the Committee on the Manufacture of Compressed Gas Cylinders. This Committee recommended the following as a suitable steel for the purpose, namely:

Carbon not to exceed 0.25 per cent.
Iron not to be less than 99 per cent.
Elongation on an 8-in. test piece cut from the finished cylinder not to be less than 15 per cent.
Tensile strength ditto not to be less than 28 tons (63 000 lb.), or more than 33 tons (74 000 lb.).

128 These mild steel cylinders are annealed before being put into service and are reannealed, while in service, once every three or four years.

129 This material is in striking contrast to that used in the light weight German cylinders and in the seamless cylinders that have been generally manufactured in the United States, which material has a high elastic limit and a correspondingly low ductility. In the United States it is not customary to submit the cylinder while in service to periodic annealings. Indeed it has not even been heretofore customary for the manufacturer to anneal the finished cylinders. Average

values of the physical properties of the steel constituting these light weight cylinders would be:

Carbon, 0.55 per cent.
Elongation in 8 inches, 12 per cent.
Yield point, 55 000 lb.
Tensile strength, 95 000 lb.

130 In the United States by far the greatest number of carbonic acid cylinders thus far manufactured have been made from lap-welded bessemer steel tubes, having the following average physical properties:

Elongation in 8 inches, 22 per cent.
Yield point, 37 000 lb.
Tensile strength, 58 000 lb.

131 The British companies engaged in compressing carbonic acid for transportation point with pride to the fact that not one of their mild steel cylinders has yet exploded when in service, while no such claim can be made for the cylinders commonly manufactured either in Germany or in the United States.

132 In the United States there are no rules and regulations whatever relating to the storage and transportation of carbonic acid, either by the government or by the transportation companies. In this respect we stand in striking contrast to such nations as Great Britain, Germany, France, etc., where the transportation of carbonic acid is conducted under regulations more or less effective.

133 The only safeguarding of employees and of the public at large in the United States has been the universal practice of subjecting all newly made cylinders to a single hydrostatic test. The hydrostatic pressure commonly used is entirely sufficient, so far as mere pressure is concerned; and if this one hydrostatic test was in itself sufficient to safeguard the public at large, then every resident of the United States should feel perfectly safe when in proximity to a charged carbonic acid cylinder. But when it is considered that the success of a hydrostatic test is due to the tenacity of the material alone and not to its ductility or toughness, and could herefore be just as successfully applied to cylinders made of brittle as to those made of ductile and tough material, it becomes apparent that this test alone is not sufficient. The writer has frequently handled experimentally a glass tube containing carbonic acid under

the conditions of commercial storage for transportation, which could have withstood successfully the customary hydrostatic test to which commercial steel cylinders are subjected.

134 The customary hydrostatic test applied to compressed and liquefied gas cylinders can serve but one legitimate purpose, namely, to indicate the presence of concealed defects, such as thin places in the wall, bad welds, cracks, etc. Any attempt to apply this test to new cylinders for any other purpose will be apt to do more harm than good. Of course this statement applies only to cylinders that are to be put into service and not to those that have been selected for testing to destruction. A proper safeguarding of the public would require that this customary hydrostatic test should be coupled with a satisfactory commercial test designed to determine whether or not the material of the finished cylinder will be capable of successfully resisting the usage, or rather the ill usage, to which carbonic acid cylinders are subjected when in service. It should be recognized in this connection that any scheme of commercial tests, when applied to well designed cylinders constructed of suitable material, should aim only at indicating and eliminating the occasional cylinder or group of cylinders, which if put into service may become a menace to the public, either because of defective material or defective construction. This should be accomplished without damaging the good cylinders in any way. Any scheme that necessarily overstresses the good cylinders in order to eliminate the occasional defective ones is highly improper, notwithstanding the fact that such tests have been in common use.

135 In the body of this paper the economics of cylinder design has been fully treated from the standpoint of the physical properties of the carbonic acid. The economics of cylinder construction and maintenance yet remains to be treated. In this Addendum the situation has been sufficiently reviewed to indicate the necessity of a further investigation of the cylinder problem. It is apparent that the magnitude and importance of the compressed and liquefied gas industries, involving as they do the constant circulation of hundreds of thousands of charged cylinders, each a magazine of stored energy, are sufficient to warrant a thorough investigation, first, of the suitability of materials for cylinder construction, second, satisfactory tests for new cylinders, and third, periodic inspection, treatment, and tests of cylinders while in service. Such an investigation should be of sufficient scope to furnish data for the formulation of a set of standard specifications for compressed and liquefied gas cylinders;

which shall on the one hand guarantee for every cylinder put into service sufficient thickness of wall and toughness of material for the proper safeguard of the public, and on the other hand that neither of these shall be so excessive as to involve the unnecessary expenses due to unnecessary weight.

DISCUSSION

MR. JOHN C. MINOR, JR.[1] Professor Stewart has taken up with admirable thoroughness a most important subject hitherto practically neglected by technical authorities. The necessity of the data which he has so fully worked out, in the proper and safe design of cylinders, cannot be questioned.

2 Much credit must be given to the British Commission, which in 1896 was the first to investigate the safety of high-pressure gas-containers; they confined themselves, however, to the consideration of accidents and to drawing up specifications for standard containers, and suggesting regulations for the future conduct of the business. It does appear, as Professor Stewart says, that they were in some respects supplied with incorrect data, but their errors were in the direction of safety, and measured by results, the report of that Commission marked a tremendous advance. Its suggestions never became law, but were adopted in their entirety by the gas manufacturers, with the result, as I am informed, that there have since then been no explosions of cylinders whatever in Great Britain. Professor Stewart's investigation will, I trust, lead to an equally beneficial outcome in this country.

3 While the trade in compressed and liquefied gases has not in this country reached the proportions to which it has attained abroad, it is nevertheless of considerable volume. It is here largely confined at present to carbonic acid, with but a small production of other gases, although the oxy-acetylene welding process is rapidly bringing high-pressure oxygen into use. With Buffalo and Pittsburg as western limits, and Philadelphia on the south, there are probably 500 000 cylinders a year of all sizes circulating in the East. This industry has been going on ever since 1888, and with remarkable freedom from accident.

4 With present practice it takes 70 lb. of steel to carry 20 lb. of gas, and high rates on the railroads make the freight bill perhaps

[1] John C. Minor, Jr., Manager N. Y. Carbonic Acid Gas Co., Saratoga Springs, N. Y.

one-fourth of the whole expense. Hence temptation manifests itself in two ways: the manufacturer wants to make his tubes as light as possible in order to obtain the business, and may be prodded, if unwilling, by the buyer, who sees the possibility of reducing freight expense. The gas-producer, on his side, for every pound of gas put in the cylinder over a safe load, gets increased work and reduction of expense. I believe that the author has called a halt on all this in time to prevent the situation arising here, as in Germany, where competition carried the weight of cylinders down until the record of accidents brought governmental interference.

5 In his determination of the proportionate space occupied in a cylinder by gaseous and liquid CO_2 the author reaches conclusions whose accuracy will not, I think, be questioned. These are shown in Table 5, which well illustrates the value of this investigation, as the most recent contribution on this subject (by Lange in the Zeitschrift fur Angewandte Chemie, 1903, p. 511) gives figures at wide variance from those of Professor Stewart.

6 The author brings out the fact that with a filling of 62 per cent, i.e., with 56 lb. of CO_2 in a large cylinder, the entire cylinder volume is filled with liquid, and all supernatant vapor condensed, at temperatures between 84 and 86 deg. In England, where the regulations permit 66 lb. of CO_2 in this size of cylinder, this condition arises at temperatures above 70 deg.

7 Assuming the slight solubility of air in liquid CO_2 indicated by the author, an assumption which my own experiments do not yet tend to prove, what becomes of the air originally present in a tube thus completely full of liquid CO_2? If it is not dissolved, would the suggested allowance of 60 lb. additional pressure for each 1 per cent of air be proper in determining the actual pressure to be expected under these conditions?

8 One point in connection with the economics of design is brought out strongly if we compare our American cylinders with those of England, where a filling of 66 lb. is permitted in the size cylinder which manufacturers here recommend for 50 lb. They use a low carbon steel and put in more metal, 120 lb. I think, to our 100 lb. They thus incur a proportionately lower freight per pound of gas, and have the still greater advantage in increased storage capacity. Not much CO_2 is sold in winter, at which time the cylinders serve as storage reservoirs rather than containers for transportation. Any slight additional cost would be welcomed, therefore, which without chang-

ing the conditions of safety would insure an increase of storage capacity of 30 per cent.

9 I probably do not interpret Fig. 9 correctly, but on applying it to the two types of cylinders in common use, holding respectively 20 and 50 lb. of CO_2, the following results appear with reference to the most economic condition of storage; assuming a maximum temperature of 120 deg. the weight of the shell should be 3.05 times the weight of the contained acid, and hence the weight of a cylinder for 20 lb. of CO_2 should be 62 lb. In practice it averages 70 lb. But cylinders to hold 50 lb. of CO_2, according to Fig. 9, should take 155 lb. of metal under the same conditions, whereas these cylinders in use average 100 lb. I shall be glad to admit what I feel sure is my error, if it will aid me in interpreting one of the most important results of the investigation.

10 It is clearly brought out that protection against accident is not afforded by hydraulic test alone, on which complete reliance has hitherto been placed; provision must be made for tests of the character of the metal and its ability to stand usage. The new German regulations apply these only to new cylinders, and not to those in use, which seems essentially wrong.

11 The metal in use abroad for cylinders is so different that we cannot adopt the same regulations for testing, and much remains to be done along the lines here pointed out to provide for the safety of these containers. The question arises whether periodic renewals of the hydraulic test should not be made on all cylinders, and if so, what pressure should be set for it. Should not a certain number out of every thousand old cylinders be also subjected to an examination of their ductility, and what figure should here obtain?

12 I believe that the result of this investigation will be not only to provide data for standard specifications of new cylinders, but to induce on the part of producers of CO_2 a careful examination of all cylinders now in use. I have been assured of the concurrence of several companies in this effort to weed out dangerous tubes, and by periodic inspection and test to maintain conditions of safety, and we shall be glad to render all possible assistance, by the contribution of tubes for examination and test, etc., in any further investigation.

13 The great freedom from accident of the CO_2 industry in this country has been due mainly to our good fortune in having tubes furnished us that were better than we had reason to expect, and which have stood, almost without exception, hard usage for years. To what we have gained from good luck we may now hope to add

the advantages resulting from the application of common sense along the trail blazed by the author.

Mr. HERMAN E. STÜRCKE.[1] In a general way, the scientific part of this paper may be accepted as substantially correct. The writer would like to call attention, however, to the figures of Thiele and Deckert (Zeitschr. für angew. Chemie, 1907), showing that 1 per cent of foreign gases causes a pressure increase of about 75 lb., as against *60 lb.*, as stated. The author may also learn there of a possible error in some of his experiments. His method of determination of foreign gases at a temperature above the critical point materially facilitates the solution of the problem of the influence of foreign gases.

2 As a manufacturer of carbonic acid, the writer welcomes the author's investigation, but does not agree with his conclusions and recommendations. It is not clear to him why the author should come to his conclusions only now, after having the data for over four years. Knowing my fellow-beings to be in imminent danger, and in ignorance thereof, I am either very careless in fulfilling my duty, if I postpone the warning cry, or else the danger is more imaginary than real.

3 On May 26, 1904, at a meeting of carbonic acid manufacturers the author stated that he had made for a cylinder manufacturer a very thorough inquiry into the physical properties of CO_2 as bearing upon the questions of storage and transportation, and that, while the investigations had not yet been completed and formally reported to the cylinder manufacturer, he had been granted the privilege of submitting the tables and charts then ready. The author at this time stated his belief that the standard cylinders of that company were entirely safe, with the most economic filling at 62 per cent, at a temperature not to exceed 110 deg. fahr., and at a maximum of 3 per cent of air present.

4 The author also gave as the factors of safety, for seamless cylinders

at 110 deg. fahr., for 55 per cent, 4.1; for 66 per cent, 3.4
at 130 deg. fahr., for 55 per cent, 3.25; for 66 per cent, 2.6

for lapwelded cylinders

at 110 deg. fahr., for 55 per cent, 3.7; for 66 per cent, 3.23
at 130 deg. fahr., for 55 per cent, 2.9; for 66 per cent, 2.48

[1] Herman E. Stürcke, General Manager, Crescent Chemical Manufacturing Company, Brooklyn, N. Y.

and further stated: Seamless cylinders generally burst between 5100 and 5900 lb. pressure; lapwelded cylinders between 4900 and 5500. The manufacturers present certainly carried away the conviction that the manufacturer who had commissioned the author with the investigation was supplying safe cylinders. He should enlighten us as to why he now differs from this opinion.

5 In 1904 the author knew of only five explosions of CO_2 cylinders. The writer knows of only two explosions in the United States since that date and both were caused by overloading and undue exposure to heat or excessive shock. During the 24 years of the existence of the CO_2 industry in the United States, seven or possibly eight cylinders have exploded, with a loss of two lives: on an average one explosion in three years, or about one cylinder in 5 000-000 cylinders filled. This estimate does not, of course, include explosions caused by fire, or explosions of soda-water tanks and carbonating apparatus, frequently reported in the press as explosions of carbonic acid cylinders.

6 This very small proportion of explosions is likely to decrease still further on account of the gradual adoption of safety devices, and the greater care which manufacturers take to prevent overloading. The seamless cylinders, now mostly used, have a water-capacity of 88 to 92 lb., and are seldom filled with more than 50 lb. or 51 lb. of carbonic acid, giving a filling of about 55 or 56 per cent, or 10 per cent less than the 62 per cent claimed by the author as most economic, with a pressure reduction of 10 per cent and more at the danger point.

7 From personal observation and experience I know the English cylinders, of which the author speaks so highly, to be clumsy, unhandy and thick. They are, of course, safe, but their good record is partly due to their low pressure safety device. The carbonic acid industry is not so prosperous that it could carry the heavy burden of English cylinders and American customers would prefer their old system of marble dust and acid to the use of English drums.

8 As for the light German cylinders, one of the oldest American companies, which uses these exclusively, and has around 30 000 in constant service, has not lost one cylinder in twenty-two years by explosion in service or in transit. Used with reasonable care German cylinders are not as black as sometimes painted.

9 In Germany the regulations permit loading to 75 per cent water capacity. At 120 deg. fahr., a loading of 55 per cent gives a pressure of 1630 lb., 75 per cent gives a pressure of 2450 lb. If

this were reduced to 55 or 60 per cent, few explosions would be heard of, especially if safety devices should become more general. Of the eleven explosions of CO_2 cylinders in transit or in service between 1894 and 1902, in Europe, three only were caused by poor material (in 1896 and 1897); the other eight by overloading and exposure to the hot sun, hot water, a red-hot stove, etc. Explosions of German cylinders have occurred since 1902, and will continue, official testing notwithstanding. Why then impose upon the American CO_2 industry a burden which does not protect, and causes endless expense and annoyance? The disastrous consequences of a compulsory change of cylinders have been experienced in Germany. When the light steel cylinders came into use there, five companies could not survive the change.

10 Opinion has been expressed to the writer by a competent authority to the effect: "We believe as you do that with the 500 000 American-made drums in the United States, the explosions that have taken place have been infinitesimally few and we do not know of an instance of an explosion of cylinders such as you use, except on account of overloading."

11 If the author intends to secure for all interested more protection against explosion he can attain this end by a practical scheme to avoid overloading and by developing safety devices.

MR. GRAHAM CLARKE[1] Our company has been engaged in the gas business for 25 years. We have about fifty thousand cylinders in which to store and transport carbonic acid gas, nitrous oxid and oxygen; and we have never had a cylinder explode. Our cylinders are filled on an average four times a year and are shipped to all parts of the country. We have always bought our drums from reputable makers, believing it would be to their interest to safeguard their business by giving us the most suitable material and the best workmanship. While the cylinders come to us tested, we give them an additional hydraulic test in our factory.

2 While Professor Stewart thinks this test is not of great importance, we feel, from our past experience, that it has served us well. We plan to re-test our cylinders as nearly as possible at intervals of five years. We have apparatus for determining permanent stretch and test a certain number for this to see if there is any deterioration of the cylinders with age. If a cylinder shows a permanent

[1] Mr. Graham Clarke, General Manager, Lennox Chemical Co., Cleveland, O.

stretch of any extent we destroy it. When cylinders of a make unknown to us come into our factory for refilling, we have them tested.

3 In filling drums it is very important to see that they are not overloaded. Our drums are all weighed at the filling-stand, and tags giving the gross, tare and net weights are put on them. The weight of the empty cylinder is stamped on the side of the metal valve. These drums are then put in stock, and before being shipped they are reweighed. This double check serves as an extra precaution to make sure that the drums are not sent out overcharged.

4 From our long experience we believe the drums now in service are of sufficient strength, if properly tested, and not over-loaded, and while we would welcome any rules, yet we believe the manufacturers of gases, who know the practical side of business, should be consulted in regard to them.

Mr. L. H. Thullen The manufacturers of carbonic acid in this country have made numerous experiments to ascertain the pressure-rise with increase of temperature and with different degrees of filling. These figures were made for their own use, and have not been published to any extent. They have made also extensive tests of cylinders for bursting-pressure and ductility, and by chemical analysis.

2 Cylinders in general use here will contain 20 and 50 lb. of carbonic acid, and are generally 5 in. and 8 in. respectively in inside diameter, and 50 in. long. The 5-in. cylinders are generally made of $\frac{1}{4}$-in soft steel, having a tensile strength of 60 000 lb. per square inch. Some of the 8-in. cylinders are made of high grade steel of 95 000 lb. tensile strength, and have a $\frac{1}{4}$-in. wall.

3 Some of the cylinders are made in this country and a large number imported, some of high-grade steel coming from Germany. Cylinders are imported not necessarily on account of their superior quality, but on account of the difference in price between the domestic and imported article. The high price of domestic cylinders is due to the same cause that makes steel rails sell at $28 a ton in this country, while sold abroad by the same company, freight paid, for $22 a ton.

4 There are in use here by manufacturers of carbonic acid about 600 000 cylinders of different sizes. These cylinders are charged about four times a year, making a total of 2 400 000 to 3 000 000 cylinders charged a year.

5 In the last 20 years there have been, from different causes,

about half a dozen explosions of carbonic acid cylinders. Practically all these explosions were due to gross carelessness in handling and storing the cylinders, the majority being caused by the cylinders being subjected to a high temperature. Conditions that result from this latter cause may readily be seen by referring to Professor Stewart's table of pressures for different temperatures.

6 On this basis there has been about one explosion to 5 000 000 filled cylinders in the last 20 years. It is safe to assume that a larger ratio of loss of life has resulted from the overturning of teakettles.

7 The author leaves us under the impression that the proper filling for carbonic acid cylinders is about 60 to 62 per cent of their water capacity. I am of the opinion that a filling of about 55 per cent would be more correct. This would make a pressure on the walls of the cylinder of about 2000 lb. per square inch; at a temperature of 130 deg. fahr., the gas containing about 3 per cent air, which is the maximum contained in commercial carbonic acid. A temperature of 130 deg. fahr. is about the temperature that a metal object lying out in the sun would attain when the temperature in the shade is 90 deg., and as these cylinders are apt to be in the sun at some time, I think 130 deg. fahr. should be the maximum.

8 It is not logical to build the cylinder walls of sufficient thickness, or have the factor of safety sufficiently large, to meet a rise in the temperature under extraordinary conditions, such as fire, or for excessive filling. Such conditions should be cared for by means of a safety valve or safety device of some form. The means generally used is a soft metal disk, that ruptures, and exhausts the cylinder, when the pressure reaches a predetermined point below the yielding point of the metal and well below the rupturing point. This has been found to give ample protection, as demonstrated by the small number of explosions; and there is no more reason for designing a carbonic acid cylinder to meet all pressure rises than for designing a boiler to meet all increases of pressure with no safety valve.

9 I am of the opinion that the safety device on the carbonic acid cylinders should be so designed that an increase in temperature would make a decrease in the releasing point of the valve. Such a device could be in the form of a disk composed of a substance that had a yielding point at a comparatively low temperature. There are a number of substances that would fill this requirement.

10 With a safety device to release the pressure at a point corresponding to a temperature a little above the assumed maximum, it would be safe to design the cylinders with a moderate factor of

safety; 3.3 under these conditions would be ample. This would correspond to a fiber strain of 20 000 for steel having a minimum tensile strength of 65 000 lb. per square inch, and would be well below the yielding point of steel having a tensile strength of 35 000 lb. Soft steel for cylinders should have a tensile strength of not less than 65 000 lb. per square inch, a yielding point of not less than 35 000 and an elongation of not less than 25 per cent in 8 in.

11 As a 5-in. cylinder generally contains but 20 lb. of carbonic acid, and an 8-in. cylinder 50 lb., valued at 4 cents per pound, or 80 cents and $2 respectively, and the empty cylinders when made of soft steel weigh 70 and 100 lb. respectively, the freight rate being about 35 cents per hundred, it is apparent that a great effort should be made to decrease the weight of the empty cylinders. In fact, good engineering in cylinder design requires steel of the highest tensile strength consistent with reasonable ductility, and steel that will not deteriorate in physical quality while undergoing manufacture into cylinders. At the present stage of the steel industry, steel of a tensile strength as high as 100 000 lb. per square inch would be perfectly feasible for use.

12 The following would be good specifications for such steel:

Phosphorus (basic)	not to exceed	0.03	per cent
Phosphorus (acid)	" " "	0.05	" "
Sulphur	" " "	0.03	" "
Carbon	" " "	0.60	" "

Tensile strength not less than 100 000 or more than 115 000.
Elastic limit not less than 55 000 or more than 65000.
Elongation not less than 12 per cent in 8 in.

This would make a cylinder of reasonable weight, which with a calculated maximum temperature of 130 deg. and the proper pressure-releasing device would be perfectly safe with a safety factor of 3.3 or a fiber strain of 30 000 lb. per square inch.

13 With a maximum calculated pressure of 2 000 lb. per square inch, the thickness of the 5-in. and 8-in. cylinder walls would then be 0.16 in. and 0.26 in. respectively. Cylinders should not be tested to the yielding point; in fact, I would recommend a hydrostatic test at not to exceed 80 per cent of the yielding point. This test should be made only to detect any leak, imperfection of manufacture or thin spot in the cylinder.

14 I understand that in some countries it is customary to anneal the cylinders periodically; my opinion is that this would be detrimental to the cylinders and add an unnecessary burden to the manu-

facture of carbonic acid gas. Considering the small percentage of accidents in this country during the 24 years of the manufacture and transportation of carbonic acid gas, I see no reason for laying great stress on further investigation, unless it be along the line of making the cylinders lighter, and using high-grade steel, thus reducing the cost of shipment of this and other compressed gases.

Mr. E. D. Meier I want to speak simply on the question of the composition of the steel in the cylinders. I think that any comparison of the three statements in the paper is impossible, for the simple reason that they do not state the amount of phosphorus or sulphur in the steel. I speak from the standpoint of the boiler manufacturer: the problem is similar to ours except that we have to do mainly with cylinders under conditions of great heat, whereas with the carbonic acid cylinder you are dealing with cold. We know that phosphorus makes steel what is called by the mill men "cold short," that is, more likely to break or crack when cold. Sulphur makes steel "red short," that is, likely to crack when in a heated condition. I believe that the difference between English and German cylinders may be explained by the percentage of either metalloid. Then again the German cylinders of 95 000-lb. tensile strength contain a slight percentage of nickel or vanadium, which would put them in a different class.

2 I take issue with Mr. Thullen with regard to the percentage of phosphorus. Phosphorus is bad for any steel where you have much tensile stress, and there is no reason, except possibly a commercial one, for permitting more phosphorus in acid steel than in basic steel, simply because it is easier to get it out of the basic steel. The danger of an excess of phosphorus is very great. I have investigated a number of ruptures of plates that occurred after flanging, and in every single instance, taking the chemical analysis from borings near the point of rupture, it was found that the rupture was due to high phosphorus.

3 A very searching investigation was conducted by Charles L. Huston a few years ago in regard to the segregation of the metalloids. He found that in the ingot the metalloids segregated into a series of parabaloids, with the wide end at the top of the ingot, and therefore in order to make good boiler steel more must be discarded from the top of the ingot than formerly; and I do not think any specification for a vessel to contain liquid or gas under pressure is sufficient unless it gives also the percentage of phosphorus and sulphur.

MR. SANFORD A. MOSS I have been interested in comparing the pressures of saturated vapor given by Professor Stewart with those deduced from other experiments. There are some slight differences, but on the whole, the agreement is close.

2 In an investigation of experiments on vapor pressure of solid CO_2 and liquid CO_2 below 0 deg. cent.,[1] I found that the following equation gave vapor pressure of liquid CO_2

$$\text{Log}_{10}\, p = 45.8372 - 1 \div \left(0.025975 - \frac{0.4992}{T}\right) \quad (2)$$

Here p is vapor pressure in mm. Hg (760 mm. = 1 atm.), and T is absolute centigrade temperature.

3 This equation fitted observations with remarkable accuracy from pressure of about three atmospheres for super-cooled liquid at - 65 deg. to pressure of about 20 atmospheres at − 20 deg. cent. The critical point is also given by the equation.

4 I find that the vapor pressures given by Professor Stewart agree with values computed from this equation in the vicinity of the critical pressure. As the pressure becomes lower, there begins to be a slight divergence, and at 0 deg. cent. there is a difference of about 2 per cent. Below 0 deg. cent., the equation agrees almost exactly with the experiments of a number of observers.

TABLE 1 VALUE OF VAPOR PRESSURE OF CO_2 IN ATMOSPHERES (760 mm. Hg)

Degrees cent.	Computed value	Amagat's value
0	34.9	34.3
10	45.0	44.2
20	56.9	56.3
31	72.4	72.3

5 Values given by various observers for the critical point agree very closely with those of Amagat. Lowest reputable value is 31 deg.

PROF. WM. KENT Referring to Professor Stewart's statement in Par. 134, "Any scheme that necessarily overstresses the good cylinders in order to eliminate the occasional defective ones is highly improper," will a proof test of 1.5 times the working load overstress the

[1] Physical Review, vol. 26, No. 6, June 1908, p. 439.

cylinders? Suppose the yield point to be 37 000 lb. and the maximum working fibre stress 20 000 lb., what harm can be done by a proof test that stresses the metal to 30 000 lb. This would cause an elongation, elastic, not permanent, of only 0.1 per cent. And even if the elastic limit were passed and the total elongation was 1 per cent, what harm could be done, since the material has a ductility of over 20 per cent?

The Author The aim of this paper, exclusive of the Addendum, is to furnish data for the solution of problems relating to the safety and economic design of carbonic acid containers, in so far as the physical properties of the acid itself are concerned. The data given are the results of very thorough researches on the physical properties of both chemically pure and commercial carbonic acid, and it is believed are adequate for the solution of all problems that are apt to arise in this connection.

2 The aim of the Addendum is to point out the fact that the experimental data given in the paper cover only one phase of the cylinder problem, namely, from the standpoint of the physical properties of the carbonic acid itself; while other phases of the commercial problem, such as suitability of materials to cylinder construction, deterioration while in service, periodic heat treatment and tests, apparently have not yet been adequately investigated.

3 The author refrains from making a lengthy closure to this paper and discussions for the following reasons: First, the few questions raised on the paper proper will be clearly answered by a careful perusal of the paper itself. Second, the discussion of the Addendum relates chiefly to matters which apparently have not as yet been sufficiently investigated to enable anyone to arrive at definite conclusions. The author believes, after listening to the discussion of the Addendum, that there is still a necessity for a further investigation of the compressed-gas container problem.

No. 1225

THE SLIPPING POINT OF ROLLED BOILER TUBE JOINTS

By Prof. O. P. Hood, Houghton, Mich.
Member of the Society

and Prof. G. L. Christensen,[1] Houghton, Mich.
Non-Member

The object of this paper is to supply data regarding the behavior of joints made by the familiar process of rolling boiler tubes into containing holes. Articles dealing with this subject give but little information except as to the ultimate holding power of such joints, and since the joint may be condemned on account of leakage rather than for lack of maximum holding power it seems desirable to have information as to the behavior of the joint through its full range of resistance.

2 When a tube has started from its original seat the fit may be no longer continuous at all points and a leak may result although the ultimate holding power of the tube may not be impaired. A small movement of the tube under stress is then the preliminary to a possible leak and it becomes of interest to know at what stress this slipping begins. A knowledge of the slipping point of a tube in its relation to the ultimate holding power is somewhat analogous to a knowledge of the elastic limit of materials in relation to their ultimate strength, in that working stresses should be kept within the smaller values.

3 The analogy is further warranted by the appearance of the load-slip diagram from such a joint, which has a general resemblance to stress-deformation diagrams of tension tests of steel.

4 Fig. 1 is a typical diagram of the action of a 3-in. twelve-gage, Shelby cold drawn tube expanded into a straight machined hole in a 1-in. plate, the tube end projecting ½ in. and not flared. The

[1] G. L. Christensen, Assistant Professor Mechanical Engineering, Michigan College of Mines.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

figures to the left give the total force applied to pull the tube from its seat; the figures below the total slip or movement of the tube through the hole. The curve shows the relation between the load applied and the corresponding slip.

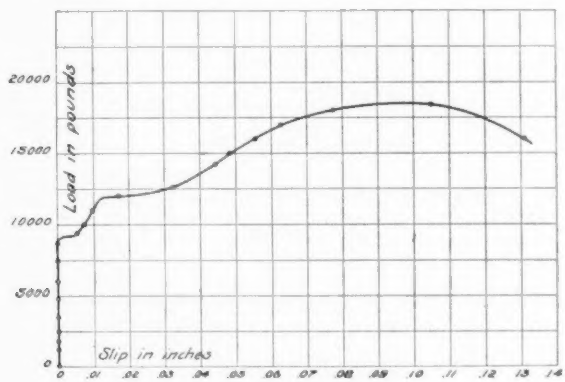


Fig. 1 Typical Load-slip Diagram

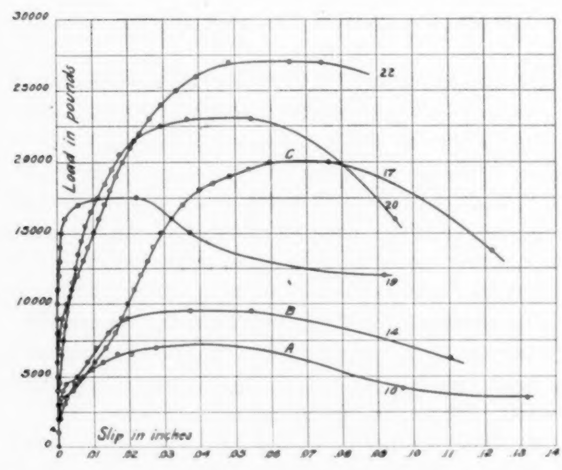


Fig. 2 Load-slip Diagrams from 3-in. Tubes Rolled into ½-in. Plates

5 The tube in this joint began to move at 9000 lb. and shows a decided slip at 12 000 lb. reaching an ultimate holding strength of 18 000 lb.

6 There is a considerable probability that this joint would leak after the tube had slipped and be condemned because of its leakiness. This slip occurs at 50 per cent of the ultimate holding strength of the joint and at 29 per cent of the elastic limit of the material in the tube.

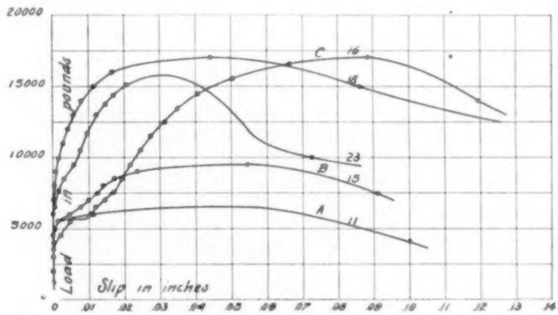


FIG. 3 LOAD-SLIP DIAGRAMS FROM 3-IN. TUBES ROLLED INTO ⅝-IN. PLATE

7 There is then a considerable field for improvement in which to raise the slipping point to a higher per cent of the ultimate strength of the joint or of the elastic limit of the tube. The usual design seems sufficient for most cases, but where high pressures are used or

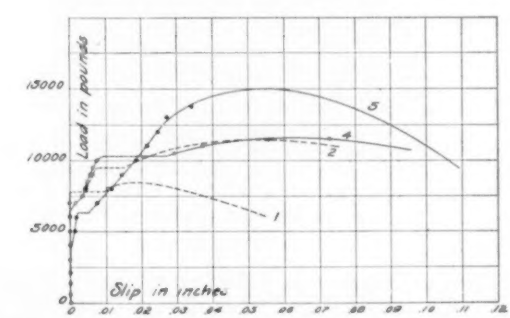


FIG. 4 COMPARISON OF LOAD-SLIP DIAGRAMS WITH STRAIGHT AND FLARED TUBE ENDS

where the stresses due to temperature variations are large, a joint with a higher initial slipping point seems necessary.

8 In many boiler designs a certain few of the tubes seem to be more highly stressed in service than others and for such designs a joint of high initial slip would be an advantage. As an illustration, a 3-in.

tube under 225 lb. boiler pressure would be urged from its seat by a force of about 1600 lb. due to pressure alone. In many tests the initial slip comes at about 6000 lb. This gives a factor of safety of 3.75 within the slipping point to take care of the unknown temperature stresses. If the design calls on the tube to act as a stay and support the pressure of but 16 sq. in. this factor of safety within the slipping point is reduced to about 1.7.

9 In attempting to strengthen the usual joint it might appear that harder rolling of the tube would raise this slipping point, but

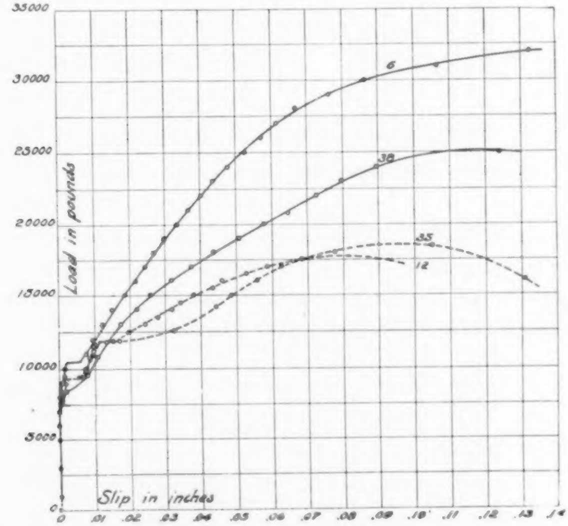


FIG. 5 COMPARISON OF LOAD-SLIP DIAGRAMS WITH TUBES IN STRAIGHT AND IN TAPERED HOLES

experiment does not show this. Harder rolling within certain limits will raise the ultimate holding power but has little effect on the initial slip. This is shown in Fig. 2 and 3, where tubes were rolled into straight machined holes in $\frac{1}{2}$ in. and $\frac{5}{8}$ in. plates. In these the tubes A,B,C were rolled respectively light, medium and heavy. Tubes A were rolled until the sheet showed a band of loosened mill scale about the hole $\frac{1}{16}$-in. wide, tubes B $\frac{1}{8}$ in. wide, and tubes C $\frac{1}{4}$-in. wide. The general agreement of the slipping points for the several degrees of rolling is noticeable although the ultimate holding power has been elevated by the harder rolling.

10 The recommendation to flare the projecting end of the tube has high authority and is of value, but while this raises the ultimate holding power it does not alter the original slipping point. It seems evident that this flared portion would have to be moved into the hole before its metal could come into play and this initial movement might be the cause of leakage. In Fig. 4 a comparison is made between tubes with straight and flared ends.

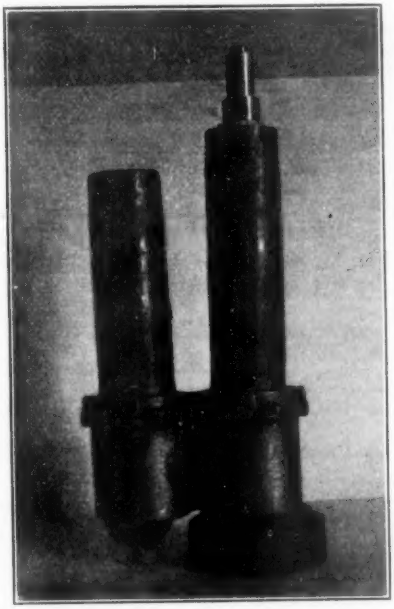


FIG. 6 TUBES EXPANDED INTO MALLEABLE IRON BOXES AND SUBJECTED TO HYDRAULIC PRESSURE

11 The ends of tubes 1 and 2 were not flared, while the ends of tubes 4 and 5 were well flared and in number 5 the hole was also slightly chamfered. Evidently the point of initial slip has not been greatly influenced by the flaring of the ends, though it is probable that a tube drawn into a hole would be less likely to leak if flared. To discover a more rigid type several forms of tube openings were tested.

12 If the holes into which the tubes are rolled are tapered $\frac{1}{16}$ in. in diameter per inch in thickness of the plate the first slipping point is hardly affected, but the joint is more rigid after a slip of 1/100 in. and the ultimate strength is increased. In Fig. 5, curves 12 and 35

represent the results from straight holes; while curves 6 and 38 are typical of those having tapered holes. These curves show the slipping points as agreeing in general, but those from tapered holes rise more rapidly and are thus more rigid.

13 During the progress of these experiments a form seemed wanted, to put the rolled metal under an initial stress in the direction of the axis of the tube, thus reinforcing the frictional resistance and making movement unnecessary to develop a larger resistance to the first slip. A tapered hole in the sheet was therefore given a reverse taper also, so that its smallest diameter was $\frac{1}{8}$ in. from the tube side of the sheet.

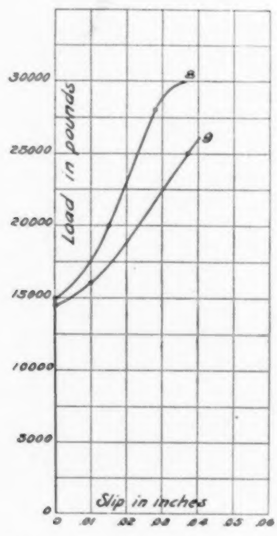


Fig. 7 Load-slip Diagrams from 3-in. Tubes Rolled into Malleable Iron Boxes and Subjected to Hydraulic Pressure

14 This amounted to a slight chamfering of the inner side of the tube sheet. Rolling the tube against the two tapers would develop such stresses along the tube as should help to resist movement. In Fig. 11 numbers 36, 44, 26, 25, 38 and 37 had double taper. Compared with the straight holes the general effect was to lower the slipping point somewhat but increase the rigidity. Two such tubes were tested by fluid pressure, the tubes having one inch bearing in malleable boxes of the form used in the Parker boiler. The combination of tubes, with closed ends, and the box, was filled with oil and an

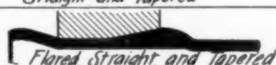
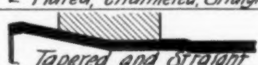
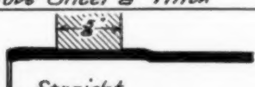
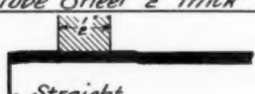
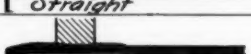
Form of Joint	Taper per inch	Test No.	Load in pounds at point of Initial Slip	Load in pounds at point of Slip = $\frac{1}{100}$ inch	Load in pounds at point of Ultimate Load	Slip in inches at point of Ult. Load
Tube Sheet 1" thick						
 Straight and Tapered		1	7800	8000	8400	.02
		2	7800	9500	11400	.05
	.06	7			17200	.12
 Flared, Straight, and Tapered	.06	4	6000	10300	11500	.07
 Flared, Chamfered, Straight, and Tapered	.06	5	4000	7600	15000	.06
 Tapered and Straight	.06	3	7000	7800	26000	.112
 Double Taper		8	15000	17500	30000	.04
	.10	9	14500	15500	21000	
Tube Sheet 7/8" thick						
 Straight		11	5000	6100	6400	.03
		15	4500	7000	9500	.04
		16	3700	5900	17000	.088
 Punched		18	8000	14700	17000	.044
 Double Taper	.10	23	6500	12000	16000	.04
Tube Sheet 1/2" thick						
 Straight		10	2000	5600	7400	.03
		14	2000	6800	9500	.035
		17	3000	6000	20000	.06
 Punched		19	10000	17200	17500	.022
		20	3500	17000	23000	.036
 Double Taper	.10	22	7000	15000	27000	.048
Tube Sheet 3/8" thick						
 Straight		13	1300	7000	18000	.045
 Punched		21	8000	15200	16500	.027

Fig. 8 Results from Tests of 3-in. 12-Gage Cold-Drawn Boiler Tubes Rolled into Various Forms of Tube Openings

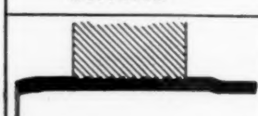
Form of Joint	Serrations: Number per inch	Serrations: Depth in inches	Test No.	Load in pounds at point of: Initial Slip	Load in pounds at point of: Slip = $\frac{1}{32}$ inch	Load in pounds at point of: Ultimate Load	Slip in inches at point of Ult. Load
Tube Sheet 1" thick							
			12	7000	11500	17700	.085
			24	6000	7000	20000	.12
			27	9000	9500	21000	.10
			32	6400	6000	6400	–
Straight Machined Hole			34	6800	10500	11500	.035
			35	8800	11200	18400	.105
Average				7333	9283	15833	.089
			25	3500	14000	23000	.08
			26	5500	12800	19700	.047
			36	8200	12600	16500	.042
			37	7500	8900	23000	.178
Double Taper			38	7000	10300	25000	.124
			44	7500	14400	33000	.069
			6	8500	12200	32000	.133
Single Taper			39	13500	17500	22600	.354
Average				7650	12837	24350	.128
	10	005	45	10000	15500	15800	.015
	10	010	46	22000	27500	27500	.008
	10	015	47	45000	50000	50000	.012
Serrated	10	020	48	43000	45000	45000	.003
	10	015	33	23000	37500	37700	.012
	10	015	40	25000	35000	35000	.01
	10	007	41	16500	23800	24200	.015
	16	007	43	21000	27200	27200	.006
Serrated	64	002	42	15000	16000	16000	.010

Fig. 9. Results from Tests of 3-in. 12-Gage Cold-Drawn Boiler Tubes Rolled into Various Forms of Tube Openings

accurately ground plunger forced in under the testing machine, as shown in Fig. 6. The results are shown in Fig. 7 as No. 9.

15 No. 8 was a tube in a similar box forced out by direct loading in the machine. The minute movement of the tubes in both tests

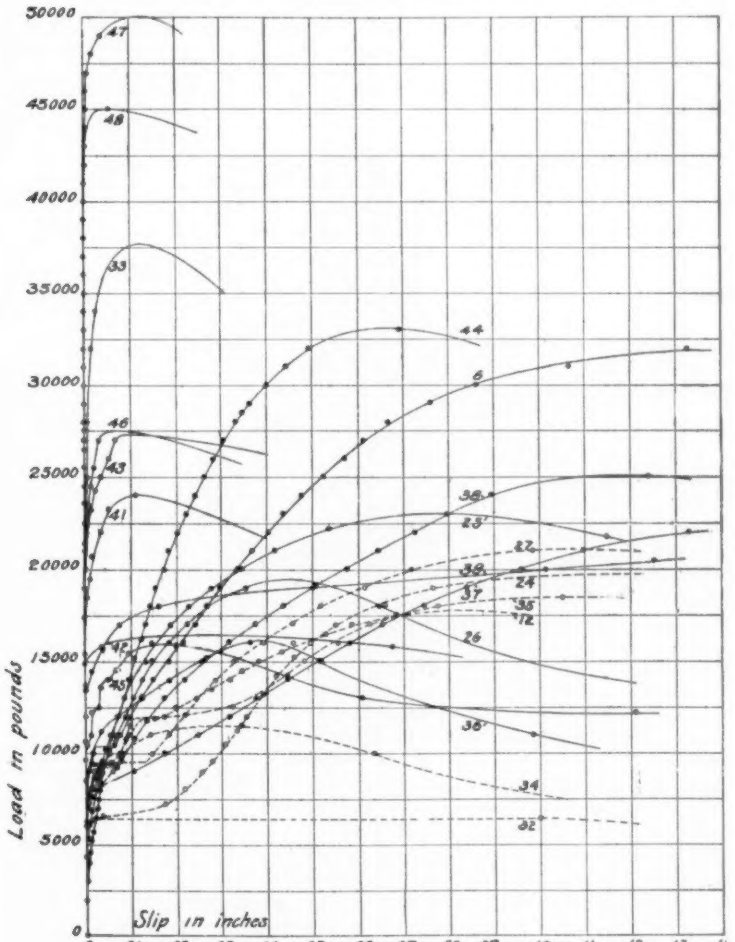


Fig. 10 Load-slip Diagrams from 3-in. Boiler Tubes Rolled in 1-in. Plates

was less easily measured in this arrangement but the eye could detect no movement of a fine scratch line at 15 000 lb. load. This corresponds to 2100 lb. per square inch hydraulic pressure in the box,

and the resistance at 1/100 in. slip was sufficient for the purposes of these experiments; this form was therefore adopted for a boiler of the Parker type designed for 300 lb. steam pressure with the feeling that the joint had as high a factor of safety within the slipping point

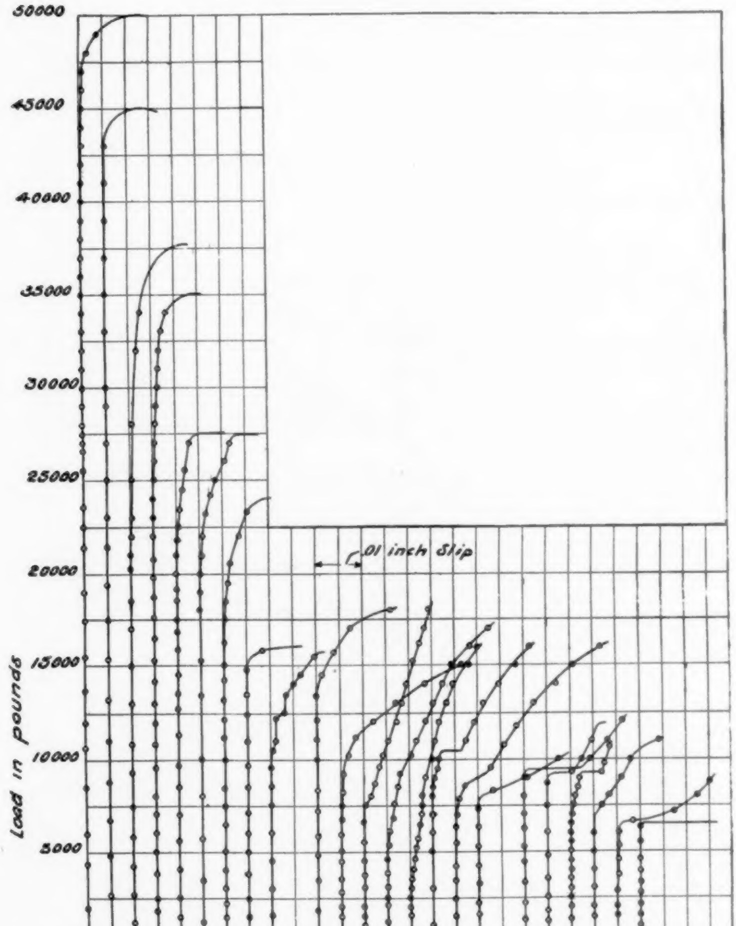


FIG. 11 LOAD-SLIP DIAGRAMS SHOWING INITIAL SLIP OF 3-IN. TUBES SHOWN IN FIG. 10, GROUPED ACCORDING TO TYPE OF TUBE HOLE

of the tube as had the rest of the boiler within the ultimate strength. Subsequent tests have shown better ways of making still stronger forms.

16 A study of the several tests made shows that in the usual machined joint the resistance to the first slipping comes from friction only. The friction is dependent on the normal pressure of the expanded tube against the sheet and this will be a maximum when the rolled metal of the tube is stressed to its elastic limit. The rolling of the metal elevates the elastic limit, but it takes a small amount of rolling to reach this maximum value. Further rolling reduces the thickness of the metal in play as fast as the elastic limit is exalted.

FIG. 12 SECTION OF TUBE NO. 47 LOADED BEYOND THE ELASTIC LIMIT

17 Assuming the elastic limit of the rolled metal at from 30 000 to 40 000 lb. the observed slipping point shows that the coefficient of friction must have been 35 to 26 per cent. The total friction per square inch of tube bearing area seems to be about 750 lb. in tube plates $\frac{5}{8}$ in. and 1 in. thick. It was observed that in straight and tapered holes wherever a high final strength was attained the metal of the tube was in some way abraded. Sometimes the sharp edge of the tube plate would shear a small ring from the metal of the tube and in other cases patches of the metal had apparently seized

and sheared. Computing the probable frictional resistance of these joints and adding the resistance of the sheared area shown on the tube gave a result agreeing closely with the observed ultimate strength of the joint as tested.

18 Most of these joints also showed a relatively high slipping point, suggesting the necessity of providing shearing resistance in addition to frictional resistance in order to obtain a high resistance to initial slip. Several forms were therefore made which provided

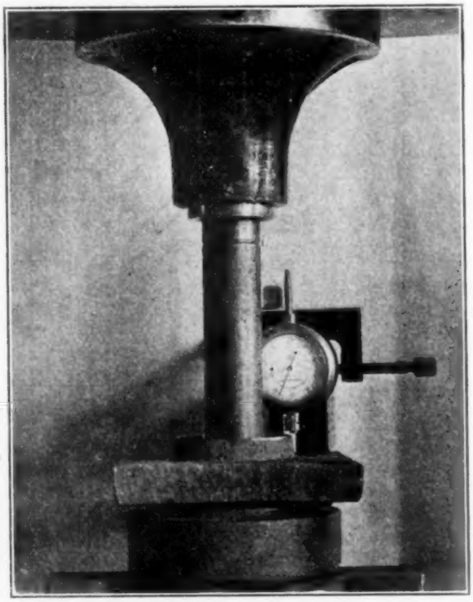


FIG. 13 METHOD OF APPLYING LOAD AND MEASURING SLIP

square shoulders in the tube sheet for the tube to be rolled against, with the object of making these several edges abrade the tube when it started to move. This serrating of the holes amounts to but little more than a "rough cut" in machining. Fig. 8 and 9 give the significant results obtained in several series of tests and Fig. 10 shows the same graphically. Fig. 11 shows the behavior of the tubes of each type up to a slight slip, thus showing the raising of the slipping point by the several methods.

19 To discover how much roughening was desirable, a series of tests were made with straight holes, in which a shallow square thread was cut with a pitch of ten threads to the inch and from 0.005 to 0.020 in. deep. The tube ends were not flared. No. 45, 46, 47, 48 (Fig. 11) show the results from these serrated holes, in which it appears that the slipping point may be very greatly elevated by this means.

20 With serrations 0.005 in. deep the surface is barely roughened and the slipping occurs at 10 000 lb. This is increased successively to 16 000, 22 000 and 45 000 lb. by increasing the depth of the grooves to 0.007, 0.010, and 0.015 in., respectively. The elastic limit of the tube is reached in tension at about 34 000 lb. and this load was exceeded by a number of the tubes before there was any slip. Fig. 12 shows a section of tube 47 which resisted 50 000 pounds before slipping. The tube was stretched 0.25 in. in a length of 3 in. and reduced 0.11 in. in diameter.

21 This figure also shows the method of applying stress to all the tubes. A plug welded into one end of the tube carried a loose hemispherical seat for the end of a central column which received and transmitted the stress from the testing machine as shown in Fig. 13. The slip was measured by means of a dial micrometer fastened to the tube sheet and arranged to measure the movement of the projecting end of the tube.

22 In test 41 the hole in the tube sheet was serrated by rolling with an ordinary flue expander, the rolls of which were grooved 0.007 deep and 10 grooves to the inch. This method of serrating is easy and can be recommended where tubes are giving trouble from slipping and are required to carry an unusual load.

23 This tube has the slipping point raised to three or four times the usual value. It appears that with serrations about 0.015 inches deep, giving an abutting area of about 1.4 sq. in. in a seat one inch wide, the maximum strength is reached as shown in tube 47.

SUMMARY

a The slipping point of a 3-in. twelve-gage Shelby cold drawn tube rolled into a straight smooth machined hole in a 1-in. sheet occurs with a pull of about 7000 lb.

b Various degrees of rolling do not greatly affect the point of initial slip.

c The frictional resistance of such tubes is about 750 lb. per square inch of tube-bearing area in sheets $\frac{5}{8}$ inch and one inch thick.

d For a higher resistance to initial slip resistance other than friction must be depended upon.

e Serrating the tube seat in a straight machined hole by rolling or cutting square edged grooves about 0.01 in. deep and ten pitch will raise the slipping point to three or four times that in a smooth hole.

f It is possible to make a rolled joint that will offer a resistance beyond the elastic limit of the tube and remain tight.

DISCUSSION

Mr. J. C. Parker Professors Hood and Christensen have shown a new phase of the subject of expanded joints. The straight non-beaded joint with about ½ in. seat has come into extensive use in water-tube boiler practice. These tests make it evident that there

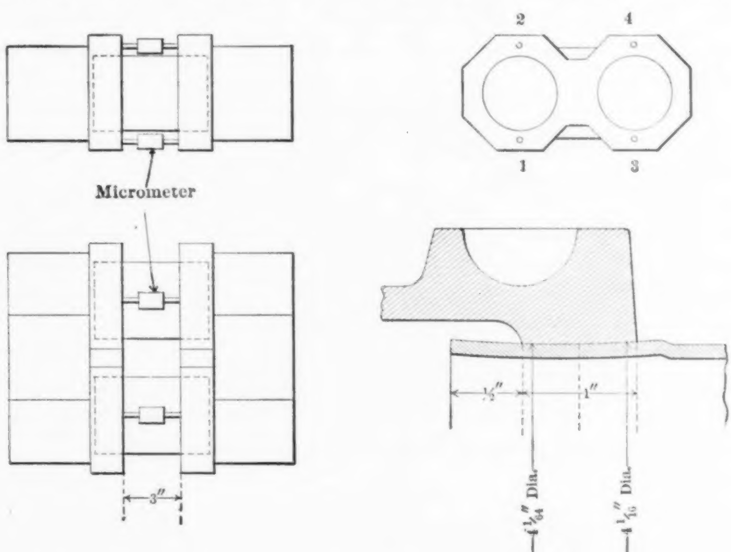


Fig. 1 Method of Measuring Slip in Tests upon Boiler Tube Joints

is considerable movement with joints of this character, especially in boilers where the tubes are not free to expand independently, and the shorter the joint the greater will be the leakage.

2 Boiler insurance companies have lately been demanding that

the tube ends project ½ in. past the seat and be flared about ⅛ in. Comparing tests No. 2 and No. 4 the flaring does not appear to have strengthened the joint in any way. This agrees with our tests, which did not show a noticeable increase in holding power from flaring over the ordinary straight rolled joint which has a slight flare from the taper mandrel. Some projection past the seat, however, is essential, and ½ in. appears to be about right.

3 When Professor Hood drew our attention to this subject we made a number of tests of 4-in. tube joints with water pressure. Pairs of boxes were connected by two nipples 6 in. long and the slip measured by a micrometer between the backs of the boxes set in

TESTS OF 4-IN. ROLLED BOILER TUBE JOINTS

Slipping point Pressure, pounds per square inch	Slip in inches				
	No. 1	No. 2	No. 3	No. 4	Average
1150	.0017	.0011	.0012	.0002	.00105
1270	.0045	.0008	.0090	.0062	.0051
1510	.0029	.0035	.0003	.0001	.0017
1650	.0024	.0023	.0008	.0012	.0017
1760	.0013	.0005	.0010	.0004	.0008
1870	.0005	.0005	.0006	.0005	.0005
1950	.0031	.0037	.0004	.0014	.00215
2060	.0006	.0012	.0016	.0010	.0011
2150	.0024	.0012	.0036	.0034	.00265
2220	.0008	.0010	.0007	.0007	.0008
2340	.0017	.0017	.0008	.0010	.0013
2450	.0023	.0024	.0011	.0012	.00175
2525	.0022	.0022	.0009	.0012	.0016
2660	.0004	.0015	.0045	.0068	.0033
	.0268	.0236	.0265	.0253	.0255

center punch marks. The accompanying table gives the results of one of the tests. The micrometer was set up in the centers so it would not spin freely and by holding it lightly between the fingers a smaller slip could be detected than could be measured. In no case did the micrometer indicate any spring or stretch or any movement whatever until the slipping point was reached, when a little jump would occur each time and all the measurements would be found to be increased.

4 The slip invariably began around a pressure of 1150 lb. per sq. in. and further slip occurred generally with each 100 lb. added to the pressure. A pinhole in a handhole cover developed at a pressure

of 2100 lb. per sq. in. and was calked. At 2660 lb. one of the tube joints began to leak.

5 Short joints are likely to leak from slipping, but a tube well rolled in a long joint acts like a tight-fitting piston which can slip without leaking. The joint shown has proved in practice to have ample holding power and rarely shows any leakage with ordinary rolling.

Prof. C. H. Benjamin Professors Hood and Christensen are to be congratulated on the success of their experiments in a comparatively new field. Their work seems to offer the first reliable data on the holding power of expanded boiler tubes, a subject on which there has always been considerable discussion and speculation.

2 The use of this method of fastening tubes in modern sectional boilers, where the integrity of the boiler depends upon the holding power of the tube, has rendered the question of even more importance than formerly. In the water-tube boiler of the Babcock & Wilcox type, where the front headers connect with the steam drums above and the rear ones with the mud drum below, the safety of the structure depends entirely upon the permanence of the union between an expanded tube and the header or drum.

3 It seems to me there are one or two points brought out in these experiments which are particularly interesting. First, the slight slipping of the joint at a comparatively low pressure and the fact that this slipping is not particularly affected by increased rolling or by flaring the ends of the tubes. Although it may be desirable to raise this slipping point somewhat, as a matter of economy, it is undoubtedly safer to have it stay where it is. Since slipping will usually be evidenced by leakage, we have here a warning of failure at a comparatively good factor of safety. If the slipping pressure is brought too near the breaking pressure, the first evidence of weakness might be complete rupture of the joint.

4 Second, that flaring does not seem to have the important effect as a safeguard which has sometimes been claimed. There has been considerable argument in the past, especially between boiler makers and insurance companies, as to the necessity or advisability of flaring. It is evident from the diagrams, as in Fig. 1, that initial slipping will not be prevented by flaring to any extent and, as in the case of Curves 2 and 4, the flaring would have but little effect on the ultimate strength. All of the diagrams and figures would seem to show that medium or hard rolling has more effect in raising the ulti-

mate strength of the joint than any flaring of the tube. I have always believed this to be the case, because rolling produces the friction between shell and tube to prevent initial slipping and it also produces a shoulder or abutment outside the shell which comes into play a little later (See Fig. 1). The flaring, on the other hand, does not come into play until there has been considerable slipping of the tube. The claims made by some authorities that flaring increases the strength 300 per cent are evidently erroneous. The difference is probably due to better rolling.

5 The margin of safety in expanded tube joints can be illustrated by considering a 4 in. boiler tube of 10 gage, expanded in a plate $\frac{5}{8}$ in. thick. The net area of the tube will be 1.627 sq. in. and if we call its tensile strength 60 000 lb. per sq. in. the ultimate strength of the tube in tension will be 97 620 lb. Allowing for friction between tube and shell 750 lb. per sq. in., it would require about 5900 lb. to slip the joint and from 8000 to 12 000 lb. to pull it out. With a pressure of 150 lb. per sq. in. inside the shell, the pressure tending to push out the tube would be 1884 lb. We thus have a factor of safety of over 3 as regards initial slipping, and as regards ultimate failure, a factor of safety of from 4 to 6. A comparison of these figures, however, with those representing the strength of the tube, will show that we are very far from having 100 per cent joint.

6 The facts brought out with regard to the serrating of the surfaces are very interesting. I think this should be done with care since it is better to have the point where leakage will show considerably below the ultimate strength of the joint.

Mr. E. D. Meier I think the tests including even the least result show that there is ample security against danger from slipping of the tubes. The authors have shown by their careful experiments that the opinion prevalent among boiler manufacturers that it is not necessary to flare the tubes if you roll them in tight, and further, that it is not necessary to use excessive rolling, is well grounded.

2 A simpler method than that suggested for preventing leaky tubes is to keep oil out of the boiler. I have never known a case of leakage around tubes that had been well rolled and properly set that could not be traced to oil. If in rolling in the tubes there is the slightest film of oil left on the tube or sheet there will be the same trouble.

Mr. M. W. Sewall I have been interested in comparing the slipping point referred to in the paper and the leakage point as shown in the tests on tubes, made by the Babcock & Wilcox Company.

2 The tests referred to were made by subjecting the tubes to hydrostatic pressure after they had been expanded into properly prepared flanges, having ½ in. wide seats. The fluid pressures at the leakage points are given below, whereas the tables in the paper give the force tending to drive the tubes out of their seats. With very light and improper expanding on 3¼ in. tubes, leakage occurred in one case at 300 lb. pressure, in another at 500 lb. per sq. in. Reduced to the stress tending to force the tubes out of their seats this becomes 2490 and 4150 lb. respectively.

3 When expanded properly the leakage points for 3¼ in. tubes were 800 to 1450 lb. per sq. in. These become, when reduced to the stress tending to force the tubes out of their seats, 6640 and 12 035 lb. respectively. The end pressures at the leaking point are somewhat comparable with those shown by the curves in Fig. 2 of the paper. It will be noticed, however, that the values obtained with the light and improper expanding, namely 2490 and 4150, are somewhat higher than those showing the slipping point in the curves A, B and C. The figures for the end pressure in the cases of properly expanded tubes, which show 6640 and 12 035 lb. respectively, are higher than the initial slipping point shown in curves 22 and 19. In fact, there is no case in Fig. 1 to Fig. 5 showing an initial slip as high as 12 000 lb. There are four curves, three in Fig. 2 and one in Fig. 3, showing slips of 0.01 in. with a stress higher than 12 000 lb.; all the others are considerably lower.

4 I would like to ask whether, where a decided thickening of the tubes is shown in the tube sheet in Fig. 8 and Fig. 9 there was an error in making the cuts. Apparently not, because it is repeated several times. I would like to know how the thickening is accomplished if the illustrations correctly represent the conditions.

The Authors In order that comparisons can be made with tests upon other tubes it is necessary to note that the radial pressure and therefore the friction which can be developed by the rolling process varies directly with the thickness and inversely as the diameter of the tube.

2 Also a tube of larger diameter and the same normal pressure, having a greater circumference and bearing area, would offer a greater resistance to slip in proportion to the increase in diameter.

3 Also if the force be applied as an internal fluid pressure this is added to the normal pressure produced by rolling and should increase the holding power.

4 Mr. Sewall's direct experiments on the leakage point as distinguished from the slipping point are of great interest and value, and it is hoped that the complete experiments may be made available to all. The figures which he has given should be modified to make them comparable with the three inch tube tests.

5 The $3\frac{1}{4}$ inch tubes used by the Babcock & Wilcox Co. are presumably standard 11 gage tubes which are 10 per cent thicker than the 12 gage 3-inch tubes. The tube diameter is also $8\frac{1}{3}$ per cent greater; therefore the normal pressure probably developed in the $3\frac{1}{4}$-inch tube when rolled would be 110 per cent $\div$ 108.3 per cent $=$ 101.5 per cent of that in a 3-inch tube. The area of the seat would also be $8\frac{1}{3}$ per cent more than in the 3-inch tube so that we should expect $101.5 \times 108.3 = 110$ per cent more resistance to slipping if the stress were applied in the same way.

6 With the stress applied as an internal fluid pressure of 300 to 500 lb. per square inch and assuming a coefficient of friction of 0.3 then 450 to 750 lb. of the resistance found in the B & W tests with light rolling was due to the internal test pressure and the remaining 2040 to 4400 lb. was 110 per cent of what would be expected with the thinner and smaller 3-inch tubes.

7 This makes the comparable figures 1855 to 4000 lb. and quite within the range of the curves shown in Fig. 2. It would appear that had the slip of these tubes been measured it would have been found that leakage occurred with a slip of less than 1/100 of an inch, for the resistance of the 3-inch tube at 1/100 in. slip was 6000 lb. Referring to the properly expanded tubes cited by Mr. Sewall it appears that 1200 to 2175 lb. of the resistance was probably due to the friction caused by the internal fluid test pressure and of the remaining 5440 to 9860 lb., 4950 to 8960 lb. is what could be expected from a 3-inch twelve gage tube.

8 This again brings the comparable figures well within the range between curves B and C in Fig. 2. Probably the harder rolled tubes leaked before a slip of 2/100 occurred. It seems evident that direct comparison of figures should not be made from tests of 3-inch twelve gage tubes under a direct push and from tests of $3\frac{1}{4}$ inch eleven gage tubes under hydraulic pressure.

9 With proper corrections however, the two sets of experiments seem to agree very well. In Fig. 8 and Fig. 9 the cut showing the form of joint is diagrammatic only and purposely distorted to aid the eye in finding the several tests. The proper dimensions are all given so that the true form of the joint is disclosed, but had the drawing been

to scale the detail would have been too small to be distinctive. There was no thickening of the tube walls.

10 The test figures given by Mr. Parker when plotted give a load-slip curve of the same character as those shown in Fig. 10, and the values at the slipping point when corrected for thickness, diameter and fluid pressure are comparable with the values found for the same form of joint in tests 1, 2 and 4. It appears that leakage actually occurred with a slip of about 0.025 in. even with a seat one inch wide and a smooth hole. In fact this test and a reasonable inference from the B & W tests seem to show that leakage does occur with a very small disturbance of the original seating of the tube although the hole may be a smooth machined one.

11 Professor Benjamin raises a very pertinent question as to whether it is not better to have the point of weakness localized at the tube ends where leakage will give so timely a warning. While the factor of safety of the ordinary joint is ample for usual cases yet, as pointed out in the paper, there are stresses due to temperature problems not readily computed and which in some cases make a stronger joint desirable.

No. 1226

AN AVERAGING INSTRUMENT FOR POLAR DIAGRAMS

By W. F. Durand, Stanford University, Cal.

Member of the Society

Some years ago the present author called attention in a published note[1] to a form of integrating or averaging instrument for diagrams plotted in polar coördinate, to which, as is well known, the ordinary planimeter is not applicable. The application of the instrument was at that time considered only for diagrams plotted on straight radial lines, such as diagrams of crank turning effort, etc. Since that time the use of dial-recording gage instruments which trace a diagram in polar coördinates but with a curvilinear path of the tracing arm has become greatly extended, and such gages are now in common use for recording various engineering quantities, mechanical, thermal and electrical.

2 Recent discussion in the engineering press[2] has indicated a renewed interest in the question of averaging instruments for such diagrams or charts, and this fact has led to a restudy of the instrument previously described with reference to its use for all forms of diagrams of this character. Believing that the description and use of such an instrument may be of interest to the members of the Society, the present brief paper has been prepared.

GENERAL DESCRIPTION AND MODE OF USE

3 It is obviously necessary in applying an instrument to such diagrams to presuppose a uniform radial scale. This is exactly analogous to the assumption of a uniform scale for the vertical ordinate in

[1] Sibley Journal of Engineering, November 1893.

[2] Power, March 3, April 27, 1908.

Presented at the New York Meeting (December 1908) of The American Society of Mechanical Engineers.

the indicator card when averaging by the ordinary planimeter. This implies simply that equal increments in the radial distance of the tracing point from the center correspond to equal increments of pressure, or voltage, or temperature, or whatever quantity may be under measurement. The problem is then for any angle of the disk corresponding to any given period of time, to find the mean radius, and thus the mean pressure, temperature or voltage.

4 This cannot be done with the ordinary planimeter, since, as is well known, the area of the diagram in polar coördinates is proportional to the square of the radius and to the angle. By the use of the ordinary planimeter, therefore, the mean square of the radial ordinate can be found, and then the square root of this can be taken. This is

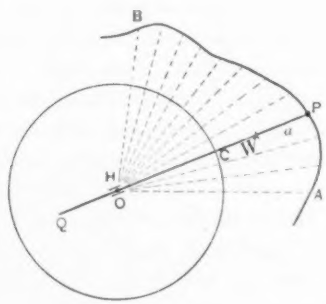


FIG. 1 DIAGRAM WITH RADIAL MOVEMENT OF TRACING POINT

not the same as the mean radius, however, a point easily proved by numerical example.

5 Consider now the arrangement in Fig. 1. Let AB denote a curve which for the present we may consider as traced by a point moving in and out on a straight radial line instead of on a curved arc as in most forms of such recording gages. Let O be the center, and at O let H denote a socket pivoted at O and permitting the rod PQ to slide freely back and forth as required to permit the tracing point P to follow the curve AB. Also let W be a wheel carried on PQ as an axis and graduated in the same general manner as the integrating wheel of a planimeter. Then it will be plain that the wheel W can respond to movement in angle only, and that movement of the rod PQ radially, or in the direction of its own length, will produce no movement and no reading of the wheel W. It is also seen that the

movement of the wheel will be proportional to the radius WO and that this differs from PO by a constant distance $PW = a$. It results that the final movement of the wheel W will be proportional to the angle moved through by the arm PQ, and to the radius OW varying from point to point along the curve. As shown in the appendix the reading for any part of the curve, as AB, is actually proportional to the product of the angle AOB and the mean radius for the curve between these points. If then we divide this reading by the angle AOB expressed in circular measure we shall have a quotient proportional to the mean radius OW. If then we add to this the constant distance WP we shall have the true mean value of the radial ordinate OP. If the curve is plotted or drawn with reference to a base circle of

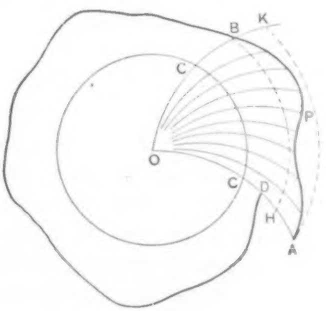


FIG. 2 DIAGRAM WITH TRACING POINT MOVING IN ARC OF CIRCLE

radius OC as datum, then the radius OC being subtracted, the remainder will be the true mean value of the ordinate CP. If WP is made equal to OC, as may readily be done by a suitable adjustment in the instrument, then the two corrections will balance and the mean value of the radial ordinate CP will be given directly as the quotient of the reading of the wheel divided by the angle AOB expressed in circular measure. If the chart corresponds to 24 hours for a complete circumference, then the angular measure to be used as divisor will be 0.2618 per hour.

6 In the usual form of instrument the marking point, instead of moving on a straight radial line as assumed in Fig. 1, moves on the arc of a circle as in Fig. 2. Then in the appendix it is shown that an instrument exactly the same as above described may be used by observing the following general instructions.

7 In using the instrument for diagrams plotted or traced as in Fig. 2 the tracing point of the instrument must start and finish at the same distance from the center of the circle. If then the diagram covers a complete revolution, as in Fig. 2, with the beginning at A and the end at D, it is only necessary to trace from D along the curved arc ODA to A, thus making the start and final finish both at A. If the diagram covers only part of a complete revolution, as for example APB, then beginning at A the tracing point is carried along APB, and then from B along the curved path BK to a point K, at the same distance from the center as A. Or otherwise beginning at B the tracing point is carried along BPA, and then along AH to a point H at the same distance from O as B.

8 In either case, the reading being taken, the mean radial ordinate is found from it by the same treatment as described above for the straight line radial path.

9 It is thus seen that the integrating or averaging instrument suited to the treatment of such diagrams is of the simplest possible form, consisting of a plain straight arm QP, Fig. 1, which serves as axis for the wheel W and slides freely through a socket H pivoted at the center of the diagram O; and that by the use of such an instrument diagrams of this character may be mechanically averaged, no matter what the actual character of the path followed by the tracing point of the recording gage, and no matter what fraction of a complete revolution the diagram may cover.

APPENDIX

10 The quantity to be determined in such diagrams is the time mean of the quantity measured by the radial ordinate. But since angular motion is made proportional to time, we may represent the desired mean by the following integral formula:

$$r = \frac{\int r d\theta}{\int d\theta} = \frac{\int r d\theta}{\theta}$$

11 Now, in Fig. 3, let $ABCD$ denote a curve drawn by a tracing point which moves on the arc of a curve shown by OAK. Then let OK, OL, OM, etc., denote a series of consecutive positions of the curve OAK, at differential angular intervals $d\theta$. Then for the actual curved path $ABCD$ substitute the broken line path made up of a series of arcs each $rd\theta$ in length, and the series of differential bits of

the curve OAK as shown. Then at the limit the record of any integrating or averaging instrument will be the same, whether the tracing point is carried along the curve or along the broken line substitute as shown.

12 Then suppose an integrating instrument, as shown in Fig. 1, applied to such a diagram, and let the tracing point P be carried along the zig-zag path. The record of the wheel will be made up of two parts:

a That due to the circular arcs $rd\theta$ and representing by summation the value of $\int rd\theta$.

b That due to the differential portions of the arc OAK.

13 Now it is clear that if the diagram extends all the way around from A through BCD to A again, the differential elements of the curve OAK may be considered as existing in pairs, and that for every

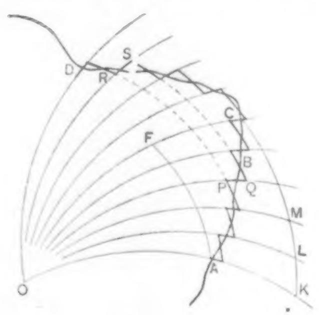


FIG. 3 CURVE TRACED BY POINTER MOVING IN ARC OF CIRCLE

element traversed in the *outward* direction, there will be an equal element traversed in the *inward* direction. PQ and RS donote the members of such a pair. The record for such a pair will therefore disappear in the summation, and hence for all the pairs, and hence for the diagram as a whole. In such case therefore part *b* above becomes O and the record of the wheel for the entire diagram consists simply of *a* or of $\int rd\theta$.

14 This reasoning is seen to be entirely general and independent of the character of the path OAK, and hence true whether it be the arc of a circle, a straight line or any other path.

15 In case the curve occupies only part of the revolution, as ABC, then it is clear that in going from A to C the record as above will involve the two parts, *a* and *b*, and that the latter will remain

included in the final result and will represent the summation of record due to the elements of OAK between A and C. It is clear that this will be the value of $\int rd\theta$ for an arc FC and hence that it will be cancelled by carrying the tracing point of the instrument back from C to F.

16 This reasoning is independent of the extent of the arc and hence equally true for an entire revolution or arc of 360 deg. where the diagram does not finish at the same radial distance as at the start, see Fig. 2. Hence in such a case it is necessary only to trace along ODA from D to A, to cancel element b above and thus to find the value of $\int rd\theta$ for the entire 360 deg.

17 It follows therefore that in all cases the correction for element b of the record is made by tracing from the terminal point of the curve along the path representing zero time change, to a point lying in a circumference passing through the initial point. Or in other words, in order to eliminate element b of the record, the tracing point must start and finish at the same distance from the center; and if the diagram does not naturally fulfil this condition then the necessary portion of the no-time-change path must be used to supplement the diagram itself.

18 It is clear that this reasoning is independent of the character of the curve OAK. It may be noted however as obvious, that if the path OAK becomes a straight line the value of the correction becomes O, and the reading of the instrument is hence independent of element b, no matter whether the diagram begins and ends at the same radial distance from the center or not.

No. 1227

NECROLOGY

WILLIAM HOLLOWAY BAILEY

William Holloway Bailey died October 4, 1908, at his residence, No. 200 West 57th Street, New York, in his seventy-fifth year. He was born in Boston, Mass., May 26, 1834. He was the pioneer of the brass and copper tube industry in this country and had been connected with the American Tube Works of Boston for over fifty-eight years and for the past fifty years was the New York representative of that company. He was the oldest member of the first panel of the sheriff's jury for the County of New York, and had been a member of the Union League Club for over forty years. Other organizations with which he was connected were the Engineers Club, Downtown Association, Society of Naval Architects and Marine Engineers, New York Yacht Club, Geographical Society, Metropolitan Museum of Art, Museum of Natural History and Academy of Design.

EDWARD BETTS BRISLEY

Edward Betts Brisley was born in New York, July 23, 1880. He attended the Dwight School in New York, and a preparatory school in Hoboken, N. J., and entered the Stevens Institute of Technology in February 1898, graduating in 1902. After leaving college he entered the employ of the Manhattan Railway Company as engineer of erection at the company's yards at 156th Street, New York. In April 1903 he was engaged by the Interborough Rapid Transit Company as inspector of construction of the subway power house at West 59th Street. A year later he accepted a position with the Crocker-Wheeler Company at Ampere, N. J., and afterward was transferred to the company's office in Pittsburg.

In February 1906 he became associated with R. M. Bailey & Co. of Philadelphia. In July of the same year Mr. Brisley became one of the partners who established the Standard Engineering Corporation, with which he was connected until his death.

Mr. Brisley was a member of the Chi Psi fraternity and of the American Institute of Electrical Engineers. His death occurred at his home at Wayne, Del., January 8, 1908.

HENRY WALLACE CAKE

Henry Wallace Cake, superintendent of the mills of the Calumet and Hecla Company, died at Lake Linden, Mich., April 21, 1908. He was born in Pottsville, Pa., March 5, 1850, and received his education in the common schools of his native state; at the age of 13 years becoming a helper in an iron foundry. Later he served three years as apprentice in a machine shop in Pottsville; worked as a machinist in a number of stationary engine and locomotive shops, and in 1887 took charge of locomotive erection in the Baldwin works, at Philadelphia; in May 1889 he became assistant superintendent of the Calumet and Hecla mills at Lake Linden, Mich., was made acting superintendent in 1901 and superintendent two years later.

GUSTAVE CANET

Honorary Member

Gustave Canet was born at Belfort, France, on September 29, 1846. He was educated at the University of Strasburg and the École Centrale des Arts et Manufactures, at Paris. He first gave attention to railway construction, and was attached to the railway works at Reichshoffen, Alsace. In August 1870, the Franco-German war having broken out, he was gazetted Lieutenant of Artillery in the Gardes Mobiles du Haut-Rhin and ordered to Neuf-Brisach with his regiment. He was present at the siege of that town, and took an active part in the construction of its lines of defense, being afterwards made a prisoner of war by the Prussian Army and sent to Leipzig. At the close of the war, and after his liberation, he resumed railway engineering work and found active employment in the construction of the Delle-Porrentruy Railway, Switzerland.

All matters relating to artillery and fortifications had always evoked in him the keenest interest, and in 1872 he severed his connection with the Swiss railway above referred to, and took an appointment with the London Ordnance Works Company, the property of M. Vavasseur, the eminent British specialist in gun construction, who at that time was established in Southwark. As early as 1876 M. Canet propounded the theory of hydraulic brakes for checking the recoil of guns, and put forward new principles for the construction of gun carriages and mountings, thus originating a new era in the manufacture of ordnance.

He left this company in 1881 and put in an ordnance department at the works of the Société Anonyme des Forges et Chantiérs de la Méditerranée, at Havre, where he remained until 1897. During that period the ordnance department of the French company in question built the armament for the foreign men-of-war designed and constructed at the shipyards of the firm located on the Mediterranean and on the Atlantic. Progress in this respect was marked, especially following upon the bill passed in the French Parliament in 1885, authorizing the free manufacture in France of war material destined for foreign governments. This led to an extension of the Havre Ordnance Works, and the construction of all types of guns, in addition to naval ordnance, was gradually taken in hand.

In 1897 the Schneider Works at Creusot was amalgamated with the Havre Works, and from 1897 to 1907 the whole was placed under the directorship of M. Canet. Vast sums were expended in developing ordnance factories and proving-grounds at both places, and a long-range proving ground was added at Tancarville.

M. Canet retired from active business at the commencement of 1907, but remained the technical adviser of the Schneider Company in matters concerning armament. At the time of his death, therefore, he had served French industry for a period of over twenty-six years.

It will be remembered that after the disastrous explosion on board the French battleship Jena, the expert knowledge of M. Canet was again called into requisition by the French Government, and he was appointed a member of a committee formed to investigate the whole question of explosives.

M. Canet was Past-President of the Société des Ingenieurs Civils de France, of the Association Amicale des Anciens Elèves de l'École Centrale, and of the Association Française pour la Protection de la Propriété Industrielle, of which he was elected Honorary President. He was, further, Honorary President of the Chambre Syndicale des Fabricants et Constructeurs de Matèriel de Guerre; Honorary Member of the Imperial Technical Society of Russia; of the Iron and Steel Institute; President of the Junior Institution of Engineers; Member of the Institution of Civil Engineers and of the Institution of Naval Architects, England; Life Member of the Imperial Institute, England; of the Naval Institute of the United States of America; and of the French Société d'Encouragement pour l'Industrie Nationale; and Life Member, Member, or Founder, of numerous other French societies and institutions. He was also Commander of the Legion of

Honor, and Officier de l'Académie, and held besides fifteen high decorations from foreign countries for services rendered them in regard to their rearmament.

M. Canet died October 7, 1908, at his seaside home, St. Aubin sur Mer, in Calvados, where since his retirement from active business he had spent a great part of his time.

KENTON CHICKERING

Kenton Chickering, Vice-President of the Oil Well Supply Company, died December 9, 1908, at his residence in Oil City, Pa. He was born in Worcester, Mass., May 16, 1847, and received his education in the Massachusetts public schools.

In 1863 he became a dispatch bearer for General Clark of the United States commissary department in New York, and remained in the Government service for a time after the war. In 1870 he represented Eaton and Cole, dealers in brass and iron goods, at Titusville, Pa., remaining with the company when it became Eaton, Cole and Burnham Company, with offices at Oil City. In 1878 Mr. Chickering was made secretary of the Oil Well Supply Company, Ltd., which was formed at this time. This new company absorbed the Eaton, Cole and Burnham Company and others. In 1891, when the Oil Well Supply Company was organized in its present corporate form, Mr. Chickering was elected vice-president, the position which he held at the time of his death.

He patented a number of useful inventions in connection with oil well machinery, and planned the large manufacturing plant erected by the company in 1901–1902, known as the Imperial Works. He also designed a number of special machines to increase the output and improve the quality of product of the plant.

Mr. Chickering was very active in church, civic and fraternal organizations.

GEORGE W. CORBIN

George W. Corbin was born in New Britain, Conn., March 3, 1859. He attended the local schools and Wilbraham Academy until 18 years of age. His first business connection was with P. & F. Corbin, who later organized the Corbin Cabinet Lock Company, of which Mr. George Corbin became manager, and later secretary and president. He resigned to become president of the Union Manufacturing Company, and held this position until his death, November 30, 1908. He

organized several other manufacturing corporations, and took an active part in municipal affairs—the savings banks, local government and schools.

He was connected with several fraternal orders, among them being the Masonic order, in which he had received the thirty-third degree, and numerous social clubs.

STERLING B. COX

Sterling B. Cox was born at Milburn, N. J., January 28, 1878, and died at East Orange, N. J., May 22, 1908. He was graduated at Harvard in 1900; was for several years with the Lackawanna Steel Company at Buffalo and New York, and subsequently secretary of the East Jersey Pipe Company.

ELMER G. EBERHARDT

Elmer G. Eberhardt of Newark, N. J., an Associate Member of the Society, died at his home on November 21, 1908. He was born in Newark, April 26, 1881, and was graduated from the Newark High School in 1896 He received his technical education in Stevens Institute and Cornell University, receiving the degree of M.E. at the latter institution in 1904.

He learned the machine trade from his father, Henry E. Eberhardt of the firm of Gould and Eberhardt, at Newark, and upon graduation from Cornell University, he formed, with his father and brothers, the firm of Eberhardt Brothers Machine Co., now the Newark Gear Cutting Machine Co. Mr. Eberhardt was vice-president of the company, and was engaged in the design of automatic gear-cutting machines, in which field he invented a number of improvements as well as investigated along original lines. He was a frequent contributor to the technical columns of the mechanical papers. He designed the power plant and equipment of the factory with which he was connected.

Mr. Eberhardt was elected president of the Cornell Society of Electrical Engineers and vice-president of the Cornell Mechanical Society, as well as receiving the honorary key of Sigma Xi for high scholarship in the scientific branches. He was an Associate Member of the American Institute of Electrical Engineers, and a member of the University Club of Newark and the Cornell Association of Northern New Jersey.

At the time immediately preceding his death, Mr. Eberhardt was engaged, aside from his business connections, in consulting engineering work, in matters relative to gears and gear cutting.

MATTHIAS NACE FORNEY

Matthias Nace Forney, whose death in this city on January 14, 1908, takes another from the ranks of the earliest members of the Society who were instrumental in its organization, was born in Hanover, York County, Pa., in March 1835.

After an ordinary schooling he was apprenticed to Mr. Ross Winans, in 1852. On the expiration of his apprenticeship he entered the employ of the Baltimore & Ohio Railroad as draftsman, serving in that capacity for three years. He then engaged in mercantile business up to the time of the breaking out of the Civil War. After this Mr. Forney filled a position as draftsman in the machinery department of the Illinois Central Railroad in Chicago. It was while engaged here that he designed what is known as the "Forney Locomotive" since used on the elevated railroads of New York. He remained with this railroad for about three years, after which he was with the Detroit Bridge and Iron Works for a short time. In the spring of 1865 he was appointed by the President of the Illinois Central Railroad Co. to superintend the building of some locomotives for that line, for the construction of which a contract had been made with the Hinckley & Williams Works in Boston. When the engines were completed, Mr. Forney remained in the employ of Hinckley & Williams, partly as draftsman and partly as traveling agent. His services here also lasted about three years. He became connected with the Railroad Gazette as associate editor in 1870. After the removal of the Railroad Gazette to New York, Mr. Forney became one of its proprietors and directed its policy on the technical side for more than twelve years, until 1884, when he retired.

He was the author of the first edition of the Car Builders' Dictionary, and the Catechism of the Locomotive.

He was an associate member of the American Railway Master Mechanics' Association, of which he was made honorary member in 1898; Master Car Builders' Association, which elected him to life membership in 1890; American Free Trade League; American Peace Society in Boston; Citizens' Union and Anti-Imperialist League of New York; and Union League, Century, Engineers' and New York Railroad Clubs.

FRED N. FOWLER

Fred N. Fowler died at Holyoke, Mass., May 16, 1908. He was born in Stratton, Vt., June 14, 1853. Mr. Fowler was self-educated, and

rose in his profession by dint of perseverance and application. For 25 years he had been associated with the United Electric Light and Power Company, of Springfield, Mass., acting as superintendent, and at the time of his death, as consulting engineer.

HARRY FRANKLIN GLENN

Harry Franklin Glenn was born at Holmesburg, Philadelphia, Pa., in 1848 and was educated in the Central High School of Philadelphia. For some years he was employed in the dry goods business in that city. In 1870, he went to Berwick, Pa., where he entered the employ of the Jackson & Woodin Manufacturing Company, car builders, holding successively the positions of clerk, superintendent of rolling mill, treasurer, secretary, general superintendent and general manager. Upon the formation of the American Car and Foundry Company in 1899, he became assistant district manager of the Berwick Plant and was later made consulting engineer of the Eastern Division of the same company. He was the oldest official in point of service at this industrial plant, having served thirty-eight years.

Mr. Glenn became a member of the Society in 1894. He died at Berwick, Pa., September 11, 1908.

THOMAS GRAY

Dr. Thomas Gray, Vice-President and Professor of Dynamic and Electrical Engineering of Rose Polytechnic Institute, Terre Haute, Ind., died December 19, 1908.

He was born in Fifeshire, Scotland, February 2, 1850. He took a course in engineering at the University of Glasgow, Scotland, where he graduated in 1878 with the degree of B.S., and shortly afterwards gained the Cleland gold medal of the University for a thesis on "An Experimental Determination of Magnetic Moments in Absolute Measure." Later he took a four-year course in practical physics and telegraph engineering under Sir William Thomson (Lord Kelvin).

In 1878 he entered the service of the Japanese government as demonstrator of physics and instructor in telegraphy in the Imperial College of Engineering, Tokio. While in Japan, he interested himself in seismographic investigations, writing a number of papers on the subject, and inventing several forms of apparatus for the observation of earthquake phenomena. After this engagement, 1881, he was employed by Sir William Thomson and Professor Fleming Jenkin, engineers of the Commercial Cable Co., to superintend the manufac-

ture and the laying of that Company's system of transatlantic and other cables, and had sole charge under them, as resident engineer, of the whole of that work. He was later chief assistant to Lord Kelvin in his engineering work. It was during this time that Lord Kelvin invented and patented his electric balances, and Dr. Gray helped him in all his experiments, both in the University and in the workshops of James White & Co., Glasgow. He also carried out several physical investigations, notably a long series of experiments on electrolysis of silver and copper solutions, and was the author of several papers on the subject of electrical measurement, which appeared in scientific journals.

In 1888, he was appointed to the professorship at Rose Polytechnic Institute, and held the position until his death.

Doctor Gray was the author of "Directions for Seismological Observations," in the British Admiralty Manual of Scientific Inquiry; of articles on telephones and telegraphs in the Encyclopædia Britannica, and of the Smithsonian Physical Tables. He also wrote many papers on scientific and technical subjects, and was engaged as an expert in electricity on the staff of the Century Dictionary. He was called upon for a great deal of outside expert work, acting as commissioner for technical and State inquiries on several occasions.

GEORGE WARREN HAMMOND

George Warren Hammond was born at Grafton, Massachusetts, April 4, 1833, and died at Yarmouth, Maine, January 6, 1908.

He was educated at Cambridge, Massachusetts, and in 1900 received the honorary degree of A.M. from Bowdoin College.

He began his career in a mercantile establishment at Long Wharf, Boston, and was subsequently employed in a wholesale dry goods store. In 1854 he went to the Cumberland Paper Mills near Portland, Maine, and after a few years became manager. During this time he increased the capacity of the works many times, making a large success of the enterprise.

He resigned from the Cumberland Mills and entered the Massachusetts Institute of Technology as a special student on the chemistry of paper manufacture. In 1876 be became manager of the Forest Paper Company at Yarmouth, Maine. This mill was a pioneer in the manufacture of soda pulp and Mr. Hammond introduced many economies in this line of paper manufacture.

He retired from active business January 1, 1906.

Mr. Hammond had great ability as an organizer of industrial establishments, was a strong believer in the training of young men, and was one of the originators of what is now known as "welfare work" which provides the most favorable conditions for the housing and care of his help.

He served in the Maine Legislature from 1868 to 1870, was a member of the Maine Board of Agriculture, and also took active part in the collection and publication of Maine vital statistics.

He was interested in botany and mineralogy and was a member of the visiting committee of the botanic gardens and herbarium of Harvard University from 1888 until his death. He was a member of the Congregational Church at Grafton and of Trinity (Episcopal) Church of Boston.

In educational matters, he was trustee of the Gorham Academy, president of the board of trustees of the North Yarmouth Academy, chairman of the trustees of the Merrill Memorial Library at Yarmouth, Maine, and trustee of the Thatcher School Associates, Westbrook, Maine. He held numerous local offices in the two Maine towns where he lived. He was a 32d-degree Mason, and in addition to membership in this Society, was a member of the American Pulp and Paper Association, American Association for the Advancement of Science, Society of Chemical Industry, London, American Institute of Mining Engineers, Society of Arts, Boston, Franklin Institute, Philadelphia, Massachusetts Historical Society, Horticultural Society, New England Historical and Genealogical Society, Bostonian Society.

While a man of force in the administration of the great manufacturing responsibilities under his charge, his personal life was that of a student, and he devoted himself to study and special investigations.

FRIEDRICH GUSTAV HERRMANN

Honorary Member

Friedrich Gustav Herrmann, Privy Councilor of the German Government and Professor of Mechanical Engineering at the Technical High School of Aachen, died on June 13, 1907. He is revered by all who know his work as teacher, scholar and engineer.

Professor Herrmann was born December 19, 1836, at Halle a. d. S., the son of a saddler. At the completion of his elementary education, he attended the first class of the provincial trade school of his native city, leaving in 1854 with testimonials of fitness and with the predi-

cate "passed with distinction." From 1855 to March 1859, he studied at the Royal Trade Institute in Berlin. After he left the High School he was employed as civil engineer in Berlin until 1868, when he entered the fourth deputation (patent board) of the Prussian ministry of commerce as assistant. But he soon turned to teaching, continuing in the work until April 1, 1906, when he was pensioned, at his own request, on account of ill health.

In the fall of 1868, Professor Herrmann accepted a position as tutor in the Royal Academy of Building at Berlin. At the same time, from 1869, he was assistant to the General-Director of Telegraphs and also acted as expert in the Royal City Court of Berlin.

The year 1870 marked a change in Prof. Herrmann's professional activity. He accepted a position in the Technical High School at Aachen, as teacher in ordinary of mechanical technology. In the year 1872 he received the title of professor.

Versatile in speech, clear and precise in expression, master of his science, Professor Herrmann was one of the best teachers at Aachen.

His great life work was the continuation, or more strictly speaking, the revision of *Weisbach's Lehrbuch der Ingenieur und Maschinen-Mechanik.* Of this work he himself says, in the preface to the first published volume 1875:

> Through the publishers the honor was offered me of continuing the publication of the fifth edition of *Weisbach's Lehrbuch der Ingenieur und Maschinen-Mechanik*, interrupted by the death of the author. If I accepted, it was not without being aware of the great difficulties of such an undertaking, or without the consciousness that I must apply all my strength and assiduity to the work, in order to any extent to justify its assignment to me. How far I have accomplished this last, I must leave to the judgment of the reader. I can at least assure him that I have not failed in application.

No one who looks in even a cursory manner at that work of seven volumes, the last of which appeared in 1901, will need any other assurance of Professor Herrmann's industry and scientific knowledge. With a wonderfully comprehensive knowledge of literature, he knew how to present with fine clearness and simplicity things already known in a new way, and the new results of his own research.

The latter are recorded in a number of works the greater part of which appeared in technical journals, although in some cases his work was presented in book form. His scientific activity was appreciated beyond the sway of the German language and translations of some of his chief works were made into English and other languages.

Professor Herrmann was very active as a member of the *Verein*

deutscher Ingenieure. Among the numerous honors which he received was that of privy councilor to the government and the honorary degree of Doctor of Engineering from the Technical High School of Karlsruhe. Dr. Hermann was the first Honorary Member of this Society, elected in 1884.

WARREN E. HILL

Warren E. Hill was born in New York in 1835. In 1852 he entered the service of the Allaire Iron Works in Newark, N. J., and was associated with that company for six years. In 1858 he was appointed superintendent in charge of the installation of the Detroit (Mich.) water works, which position he held until 1862, when he returned to the East and accepted a position with the Continental Iron Works of Brooklyn, N. Y. In 1888 he was made vice-president, and in 1907 president of this firm, the position he held at the time of his death. Mr. Hill was the designer of the machinery and engines of the original "Monitor," which defeated the "Merrimac" in Hampton Roads.

His death occurred in New York, December 8, 1908. He became a member of the Society in 1884.

WILLIAM HIGGINS HUME

William Higgins Hume was born in Polmont, Sterlingshire, Scotland, on July 12, 1877. He came to America with his parents in 1883, residing a few years in South Carolina. Later they removed to Troy, Ala., where at the Troy Normal College Mr. Hume received a part of his education.

In 1894 he went to Scotland and studied two years in the Herriott College, Edinburgh. Upon his return to America in 1896, he entered the machine shop and office of the Standard Chemical and Oil Company of Troy, Ala. In 1898 he was draftsman with the Bethlehem, (Pa.) Steel Company and later had engagements with Jones & Laughlin in Pittsburg, Pa., and the Edgemore Iron Company at Wilmington, Del.

In 1902 he became superintendent of construction of the Georgia Iron and Coal Company at Rising Fawn, Ga., during the rebuilding of their blast furnace plant, and later was chief engineer of the same company.

In 1904 he was general sales manager of the Herron-Brady Pump and Foundry Company, Chattanooga, Tenn., leaving there in 1907 to accept the appointment of superintendent of foundry of the

Bucyrus Company of South Milwaukee, Wis. He held this position until his death, which occurred January 3, 1908.

EDWARD L. JENNINGS

Edward Lobdell Jennings, whose death occurred on November 6, 1908, was born in North Wayne, Me., April 14, 1850. He received his education in the public schools of his native town and at the Maine Wesleyan Academy. He was apprenticed to the North Wayne Tool Company, and in 1872 went to Boston and entered the employ of W. A. Wood & Co. and was their manager for several years. He resigned this position and removed to Waterbury to become purchasing agent for the American Brass Co., the position he held for the eight years preceding his death.

Mr. Jennings was a member of the Waterbury Club, a Commandery Mason and a member of the First Church (Congregational), Waterbury, Conn.

FRED. A. JOHNSON

Fred. A. Johnson, president of Gisholt Machine Company, of Madison, Wis., died in Denver, Colo., May 26, 1908. He was born in Madison, Wis., September 14, 1862, and was 46 years old at the time of his death. He attended the mechanical engineering course of the University of Wisconsin with the class of 1884, afterwards spending a number of years in the agricultural implement shops of the Fuller & Johnson Manufacturing Company, of Madison, where he rose to the position of assistant general superintendent. In 1885 he joined his father and brothers in the organization of the Gisholt Machine Company, and in 1901 became its president. Up to 1904, when illness prevented, he took an active part in the management of the business, and spent several years in Europe in the interests of the company. He was an associate member of the Institution of Mechanical Engineers.

EDWIN H. JONES

Edwin Horn Jones was born in Wilkes-Barre, Pa., April 15, 1844, and died December 2, 1908. He was educated at the Old Dow Academy on South Franklin Street, and at an early age entered the employ of his father, Richard Jones, who then conducted the Jones Foundry, the foundation of the present extensive Vulcan Iron Works. He learned the iron business thoroughly, advanced to superintendent of the works, and at the time of his father's death in 1873 became

general manager of the company and later its president, a position he held until the time of his death.

As president of the Vulcan Iron Works he consolidated the Wyoming Valley Manufacturing Co. and the Pittston Iron Works with the original plant and later purchased the Tamaqua shops, all of which he consolidated as the Vulcan Iron Works.

Nearly twelve years ago he was made president and general manager of the Sheldon Axle Works. In 1881 he became Director of the Second National Bank, and later its Vice-President. He was interested as stockholder and director in a number of other industries and was an active member of the Wilkes-Barre Board of Trade and one of its trustees. He was a member of the Westmoreland Club and the Wyoming Valley Country Club, the Arts Club of Philadelphia, and the Sons of the Revolution, and a member of the Central M. E. Church.

FRANK B. KLEINHANS

Frank B. Kleinhans was born at Easton, Pa., August 20, 1874, and died in Pittsburg, Pa., September 1, 1908. He was educated in the high school of Easton, and at Lafayette College, graduating in 1897 with the degree of Electrical Engineer, and in 1900 he was given the degree of Master of Science. In 1897 he entered the employ of the Baldwin Locomotive Works, Philadelphia, where he remained for three years. In 1900 he became associated with Bement, Niles and Company, Philadelphia, as designer; in 1902 he became chief draftsman of Lodge & Shipley Machine Tool Company, Cincinnati, O., resigning the position in 1903 to become chief engineer of the Fischer Foundry and Machine Company of Pittsburg; he left this company in 1906 for the United Engineering Company, with which firm he was connected until his death.

He wrote extensively for the technical journals and was the author of a book on Boiler Construction. At the time of his death, he was writing a series of articles for The Boiler Maker on the subject of Flanging Boiler Plates.

He was a member of St. John's Lodge, No. 219, Free and Accepted Masons, Pittsburg, Pa.

ALBERT FRANKLIN KNIGHT

Albert Franklin Knight was born April 10, 1854, at East Greenwich, Rhode Island, and attended schools at Bristol and Providence.

From 1868 to 1875 he was an apprentice to the John L. Ross cotton mill, in the repair shop at Eagleville, Connecticut, and was made superintendent in 1875, holding the position until 1879. He was with B. B. and R. Knight, Providence, from 1879 to 1880; with the Bozrahville Company, Bozrahville, Connecticut, from 1880 to 1888, in both places acting in the capacity of superintendent of cotton mills. He built additional buildings and placed machinery, engines, boilers, etc. He was manager of the Canada Cotton Company, Cornwell, Ontario, from 1888 to 1889; superintendent of the Lonsdale Company, Ashton and Lonsdale, Rhode Island, from 1889 to 1891; agent of the Amory Manufacturing Company, Manchester, New Hampshire, 1891 to 1898.

From 1898 to 1900, Mr. Knight was agent of the Berkshire Manufacturing Company of Adams, Massachusetts. Leaving there he entered business for himself, leasing and operating the Ray Cotton Mill at Woonsocket, R. I. Upon the expiration of his lease, he purchased the Farmers Cotton Mill at Farmesville, Massachusetts, and moved the machinery to a mill in Huntsville, Alabama. Afterward he engaged in the buying and selling of machinery and mill properties under the name of Providence Machinery Exchange. He was manager of this company at the time of his death, February 4, 1908.

FRANCIS X. McGOWAN

Francis X. McGowan, whose death resulted from a railroad accident on September 4, 1908, was born in Lawrence, Mass., February 11, 1877. He attended the public schools of that city, graduating from the high school in 1895, after which he entered the Massachusetts Institute of Technology, graduating from the mechanical course with the class of 1900. After graduation he entered the power division of the Western Electric Company, New York, and was later transferred to the engineering sales department. He remained in the employ of this firm to the time of his death.

Mr. McGowan was a member of the National Geographical Society and became a Junior member of this Society in 1902.

OTHNIEL FOSTER NICHOLS

Othniel Foster Nichols, consulting engineer for the Department of Bridges, New York, died February 4, 1908. He was born in Newport, Rhode Island, July 29, 1845. After preliminary training in the New York public schools he entered the Rensselaer Polytechnic Institute and was graduated as a civil engineer in 1868. He was immediately em-

ployed as assistant engineer in laying out Prospect Park, Brooklyn, and in 1869 was appointed junior assistant in the construction of the first elevated railway in New York. In 1870 he became assistant engineer in the office of Cooper and Hewitt. The following year he went to Peru to construct the tunnel division of the Chimbote Railroad, and continued this work for four years. In 1878, after his return to the United States, he served as assistant engineer and superintendent for the Edge Moor Bridge Works, in the construction of the Metropolitan Railway, and was employed by the city as engineer in charge of the main drainage sewer for the annexed district. He then went to Brazil as resident engineer and attorney for the Madeira and Mamore Railway. He returned to the United States and reëntered the employ of the Cooper and Hewitt Company, serving two years as assistant engineer in the shops of the New Jersey Steel and Iron Company, and later as assistant to the president of the Peter Cooper Glue Factory. Subsequently he was appointed resident engineer of the Henderson bridge over the Ohio river and later, chief engineer of the Water Works Company of Westerly, Rhode Island. He resigned that position to become chief assistant engineer of the Suburban Rapid Transit Company of New York. Afterward he became chief engineer of the Brooklyn Elevated Railroad and also acted in that capacity for the Railway Construction Company and the Union and Seaside Elevated Railroad Companies. In 1892 and 1893 he was general manager of the Brooklyn Elevated Railroad. In 1895 he was appointed assistant engineer in the planning of the Williamsburg bridge, which office he held until 1903. He was then appointed assistant engineer in the Department of Bridges, and in 1904 was made chief engineer of the department, later acting as consulting engineer.

Mr. Nichols was a member of the American Society of Civil Engineers, the American Geographical Society, the British Institution of Civil Engineers, the Engineers' Club, president of the engineering department of the Brooklyn Institute and a trustee of the Brooklyn Collegiate and Polytechnic Institute.

WILLIAM A. PEARSON

William A. Pearson, for 16 years chief architect and superintendent of buildings for the General Electric Company, at Schenectady, N. Y., died in that city May 26, 1908. He was born at Athens, Bradford county, Pa., July 29, 1855. After graduation from the Sayre, Pa., high school in 1870, he served his apprenticeship to the machinist's trade in the D., L. & W. Railroad shops in Scranton, Pa.,

where he was foreman two years. He was journeyman machinist and locomotive engineer with the Union Pacific Railroad in Omaha, Neb., and afterwards foreman of shops at Carson City, Nev. He then went to Virginia City, Nev., where he became superintendent of the Comstock mines, but resigned to engage in mining business with headquarters in New York. He afterwards became superintendent of the marine department of the Dickson Manufacturing Company in Scranton, later occupying a similar position with the Boies Wheel Company in the same city. His connection with the General Electric Company followed.

FREDERIC A. C. PERRINE

On October 20, 1908, Frederic Auten Combs Perrine died at his home in Plainfield, New Jersey, thus bringing to an end a life of usefulness hardly more than well begun.

Dr. Perrine was born at Manalapan, New Jersey, August 25, 1862. He was prepared for college at the Freehold Institute, New Jersey, and entered Princeton University in 1879. He received from that institution the degree of A.B. in 1883 and remained as a post-graduate until 1885, when the degree of Sc.D. was conferred upon him, and he was nominated one of the honor men of that year. In 1886 Princeton conferred upon him the further degree of A.M.

The habit of study, acquired at college, lasted until shortly before his death, his interests constantly broadening. He specialized very early in electrical engineering and on leaving college associated himself with the United States Electric Lighting Company of New York, one of the first electric manufacturing and operating companies in this country. Dr. Perrine early developed an interest in matters other than engineering, and for many years devoted considerable time to university settlement and other sociological work, his active interest relaxing only when his professional duties became too exacting.

In 1889 he became associated with John A. Roebling's Sons Company as manager of the insulated wire department, retaining this position until 1892, during which time he undertook a great deal of investigation and research, with a view to placing the manufacture of insulated wire upon a scientific basis. He published the well-known "Roebling Handbook," which has since gone through many revisions and editions, but which, at the time of its first publication, was practically the only comprehensive handbook of the character in this country.

His investigations while in the wire business developed an interest in conducting materials which lasted until his death, and resulted in the publication of his book on "Conductors for Electrical Distribution" in 1903.

In 1892 he became associated with the Germania Electric Company of Boston as treasurer and manager. His connection with this company was brief, which was probably fortunate as it left him free to enter into his happiest and most successful work—that of teaching. In 1893 he was appointed Professor of Electrical Engineering at Stanford University in California. Prior to this time he served as judge and member of several important committees of the World's Fair at Chicago.

At the time of Dr. Perrine's appointment at Stanford University, the electrical engineering department had not been organized. He installed a course which successfully combined the study of the physics and mathematics of electricity and its application to engineering, holding a balance between the theoretical and practical education.

Dr. Perrine's success as a teacher, however, was principally due to his personality and intimate contact with his men, and to his association with them outside of the lecture room and laboratory.

During the latter part of his incumbency of the chair of electrical engineering at Stanford University, he held the position of chief engineer of the Standard Electric Company of California (now a part of the system of the Pacific Gas and Electric Company), where his work included the electrical design of the system, which was the first long-distance 60-kilovolt transmission plant to be undertaken. He was awarded a gold medal at the Paris Exposition in 1900 for this work.

In 1900 he became president and general manager of the Stanley Electric Manufacturing Company of Pittsfield, Mass., a position which he occupied until 1904, when he took up private practice as a consulting engineer in New York, which he maintained until his death.

He was a contributor to technical literature, and with Geo. P. Low was editor of the "Journal of Electricity" from its inception in 1894 until 1896, during which time his communications, under the caption of "Passing Comment," were critical commentaries on the technical affairs and developments of the times. From 1896 to 1898 he was an editor of "Electrical Engineering," published in Chicago. He presented many papers before technical societies, educational institutions and other public bodies.

He was a member of the American Institute of Electrical Engineers, serving on many important committees; of the Institution of Electri-

cal Engineers (London); the American Society of Civil Engineers; The National Electric Light Association, and other technical and learned societies.

FERDINAND PHILIPS

Ferdinand Philips was born in Elberfeld, Rhein Province, Germany, in 1850. He was educated at the Königliche Provinzial Gewerbeschule at Elberfeld, graduating with the highest honors, and at the Königliche Gewerbe-Akademie at Berlin. At Liege, Belgium, he continued his studies in French, and came to America in 1876. He was first employed by Hoopes & Townsend of Philadelphia as designer and engineer, remaining with them until 1883, when he started the wire nail industry in this country under the name of Philips, Townsend & Company.

In 1899 he invented the Philips Pressed Steel Pulley. In the latter part of 1907 the Philips, Townsend & Company and the Philips Pressed Steel Pulley Works, both of which Mr. Philips was the proprietor, were merged into one corporation under the name of Pressed Steel Pulley Works with Mr. Philips as president and general manager.

Mr. Philips died at his home in Philadelphia, March 26, 1908.

CHARLES D. PIERCE

Charles D. Pierce was born in Oswego City, New York, January 31, 1847. He was educated in the home schools of his town and served an apprenticeship in architecture and building with his father, Henry D. Pierce, from 1864 to 1868. In 1868 and 1869 he took a course in architectural and mechanical drawing in the St. Louis (Mo.) Polytechnic Institute, and served with Stacy and Stone, Architects. Afterward he went to Lawrence, Kansas, as architect and contractor, and from 1873 was engaged continuously as manufacturer of machinery for drilling artesian, oil and gas wells, and machinery for prospecting for minerals and developing mines. He was also in charge of machine shop and steam forge operations.

In 1877 he began the manufacture of well boring machinery in Philadelphia; two years later he moved to New York where he built the "Pierce Iron Works" in Long Island City, subsequently removing to Jersey City, N. J., where he owned and operated the Pierce Well Engineering & Supply Co. He patented numerous devices and appliances both in America and foreign countries.

At the time of his death he was Consul General for the Orange Free State. His work of boring artesian wells was carried on extensively in Mexico, Central and South America, the West Indies, and also in Europe, Asia and Africa.

His death occurred April 24, 1908.

JAMES POWELL

James Powell was born in Ghent, Belgium, in 1832 and his parents came to America during his childhood. In 1846 he began the work of brass manufacture at his father's factory in Cincinnati, Ohio. This business was the founding of the manufacture of plumbers' brass goods in the West.

An interesting item in the records of Mr. Powell's early business was an order from the Union Army during the Civil War for one thousand pairs of spurs of a special pattern to be delivered in 48 hours. There were no castings on hand and patterns had to be made, but the goods were shipped within two hours of the stipulated time.

In 1886 the business was merged into a stock company with Mr. Powell as president and manager, and he held this office until his death.

Mr. Powell invented many devices, among which patents were secured on globe valves, blow-off valves, lever throttle valves, improvements on lubricators, glass engine oilers, grease cups, injectors and devices for trimming engines and boilers.

Mr. Powell wrote frequently for magazines and journals, and was a reader and student.

He was a member of the National Geographic Society, the National Association of Manufacturers, Manufacturers' Club of Cincinnati, the Queen City Club; the Business Men's Club of Cincinnati and other business and philanthropic societies. He was an active worker in the Baptist Church, being a trustee and deacon for about forty years.

Mr. Powell died February 25, 1908.

EDWARD FRANKLIN SCHAEFER

Edward Franklin Schaefer, whose death occurred May 27, 1908, was born in New York, November 2, 1879. In 1900 he received the degree of B.S. from the College of the City of New York; in 1902 the degree of M.E. was conferred upon him by Cornell University and in 1903 he received the degree of M.M.E. from the same University.

He was instructor at Cornell University in connection with a fellowship under Prof. J. H. Barr and Prof. R. C. Carpenter, in 1902. From 1903 to 1906 he was in the sales and publication departments of the Ingersoll-Rand Co. In 1906 he accepted a position as chief engineer with the Poto Mines Corporation, Poto, Peru, S.A., and in 1907 he became Consulting Engineer of the Rinconada Mining Co., Poto, Peru, S. A. He held these positions up to the time of his death.

CHARLES H. L. SMITH

Charles H. L. Smith was born in New York, November 14, 1843. He attended public and private schools, and for three years was a student at Rensselaer Polytechnic Institute. He studied architecture and was engaged as architectural draftsman in the office of Thomas C. Smith, at the same time studying mechanics and machinery. In 1876 he became associated with his father in the manufacture of porcelain, the firm name being the Union Porcelain Works, located at Greenpoint, New York City, the only pottery in the United States where true hard porcelain is manufactured. It also has the distinction of being the only factory to produce hard china without the government aid received by factories abroad. During his connection with this firm Mr. Smith introduced the hard kaolinic body and invented a method which made it possible to make an oval dish by machinery. He accomplished this by applying the eccentric principle to the potter's wheel. He designed a great deal of machinery for use in his plant.

Mr. Smith was a member of the Masonic Order; life member of the New York Historical Society, the American Institute of Arts and Sciences, the New England Society; a member of the Manufacturers Association of New York, the New York Board of Trade and Transportation, the Garden City Golf Club and the Crescent Athletic Club. Mr. Smith died March 6, 1908.

RICHARD HERMAN SOULE

Richard Herman Soule was born March 4, 1849, in Boston, Mass. He graduated from the Massachusetts Institute of Technology in the class of 1872, and in 1875, entered the service of the Pennsylvania Railroad, where he remained for eight years. In 1879, he was made superintendent of motive power of the Northern Central Railway.

From October 1881 to June 1882, he was superintendent of motive power of the Philadelphia and Erie division of the Pennsylvania

Railroad, and then accepted a position in the same capacity with the Pittsburg, Cincinnati and St. Louis Railway.

In 1883, when the West Shore Railway enterprise was carried through, its managers secured the best talent available in the country for their managing officers, and Mr. Soule was appointed superintendent of motive power, a position which he held until the absorption of the West Shore Line by the New York Central in 1887. From February 1887, to April 1888, he was general manager of the New York, Lake Erie and and Western Railroad, and in November 1888, he was appointed general agent of the Union Switch and Signal Co. He was engaged in the introduction of modern interlocking and lock-signaling plants until 1891. From 1891 to 1897 he was superintendent of motive power of the Norfolk and Western Railroad, and did much to put the rolling-stock of the system, which was then coming into prominence as an important coal-carrying road, on a thoroughly sound basis.

For the next two years, Mr. Soule was in the employ of the Baldwin Locomotive Works, spending nearly a year traveling in foreign countries. He had charge of the Chicago office of this company for a year and a half.

In 1900 he opened an office in New York as a consulting mechanical engineer and practiced until, on account of ill health, he was forced to retire from active business.

Mr. Soule was a member of the Master Car Builders Association; and author of a report on the standards of this association, which led to a radical change in the association's practice, and to a placing of the standards on a much higher basis. He was also a member of the American Railway Master Mechanics Association. He was one of the managers of this Society, 1898–1901.

He was universally respected and esteemed for his many sterling qualities, which caused his acquaintance to be highly prized by his associates. In all parts of the country men are found who testify to the help given them early in life by Mr. Soule, to whom they owe much of their later success. His memory will live long in the hearts of those to whom he had endeared himself.

Mr. Soule's death occurred at his residence in Brookline, Mass., December 13, 1908.

WILLIAM DURELL STIVERS

William Durell Stivers was born in Jersey City, N. J., February 20, 1871. He received his education in the public schools, graduated

from the high school in 1887, and pursued some special studies in mechanical engineering at the Cooper Institute in New York. He entered the DeLamater Iron Works in 1887 and was assigned to special service in the superintendent's office where he had unusual opportunity for acquiring a special training in shop management, engineering and experimental work. While there he took part in various interesting experiments that were conducted at the DeLamater Iron Works, as, for instance, those of the noted Ericsson expansion engine, the hot air engine, the Belleville boiler, refrigerating and compressed air machinery, and other constructions. After the dissolution of the DeLamater Iron Works in 1889, Mr. Stivers entered the Quintard Iron Works as draftsman, eventually rising to the position of acting superintendent. He supervised the building and installation of the machinery of the U. S. S. "Maine" which was destroyed in Havana Harbor. He was the works representative on the trial trips of the U. S. S. "Concord," "Bennington," "Detroit" and "Marblehead." He also had a prominent part in the trial trip of the U. S. S. "Bancroft." In 1902 he left the Quintard Iron Works to accept the position of superintendent of the Yonkers Works of the Otis Elevator Company, which position he held until March 1904, when he was engaged by the C. W. Hunt Co. as executive engineer, remaining active in that place until his death in December 1906. He joined the Society in 1904.

JOSEPH STONE

Joseph Stone, born in Charlestown, Mass., January 4, 1848, was a member of the first class graduated from the Massachusetts Institute of Technology, graduating in 1868 with the degree of S.B. From 1868 to 1873 he was engaged as mechanical engineer in the building and remodeling of textile mills; from 1873 to 1880 he was agent of the Manchester, N. H., Mills, and from 1880 to 1887 was similarly in charge of the Lower Pacific Mills in Lawrence, Mass. He then retired from active business, devoting his time to his real estate interests.

Mr. Stone was a member of the Technology Club of Boston. He was a man of retiring disposition, but of rare business tact and sterling character. His home for the last few years was in Brookline, Mass., where he died suddenly from heart failure, May 24, 1908.

HARRIS TABOR

Harris Tabor was born in Clarence, Erie County, N. Y., January 26, 1843, and died at his home in Philadelphia, July 29, 1908,

as the result of an automobile accident which occurred July 4, 1907, in which accident Mrs. Tabor was also severely injured.

Mr. Tabor was a charter member of the Society; attended the meetings preliminary to its formal organization as well as most of those held afterward, and took an active interest in the Society's welfare.

Receiving in his native place a common school education, he supplemented this as occasion arose by study along the special lines in which he became interested. At 16 he engaged as an apprentice in the shop of his brother, Leroy Tabor, at Tioga, Pa. Two years later he enlisted in the Union army and served two years in the Civil War, after which he was employed by S. Payne at Troy, Pa., as a machinist, afterward becoming superintendent of the shops of B. W. Payne & Sons at Corning, N. Y., and later of the shops of the Hartford (Conn.) Steam Engine Company. He was then for some years with the Westinghouse Machine Company at Pittsburg. A student of steam engines, he devised various improvements upon them and his "Tabor" indicator is well known to engineers, being one of the first of such instruments capable of correctly indicating high-speed engines. In the Westinghouse establishment be became interested also in foundry matters, especially in the problems of molding by power driven machines, and became one of the highest authorities on them. He finally organized the Tabor Manufacturing Company, New York, and was its president for a number of years, but finally sold out most of his interest and the company was then removed to Philadelphia. After a few years, however, he again became active in its affairs.

Mr. Tabor devised the first successful power molding machine in which the ramming was done by an overhead inverted steam cylinder; introduced compressed air for working such machines and invented the "vibrator" for molding machines; a device which greatly facilitates the drawing of difficult patterns. Very recently he had devised important improvements and was at work upon others when disabled by the unfortunate accident which caused his death.

FREDERICK CONOVER WARMAN

Frederick Conover Warman was born January 7, 1872, at Rutherford Park, N. J. He was educated in the public and high schools of Washington, D.C., and entered Lehigh University in 1889, graduating in 1893 with the degree of C.E.

He was appointed to the U. S. Engineer Corps immediately after graduation and served in the capacities of rodman, inspector of

dredging, draftsman and transitman in charge of parties on river and harbor and bridge works. In May 1906 he was promoted to the position of assistant engineer, and at this time made designs, drawings and estimates for extensive defenses of Washington. He was placed in charge of the survey for the Memorial Bridge from Washington, D. C., to Arlington, Va., designed boring machinery and made borings, maps, and estimates for this bridge; designed, and constructed engineer wharf at Easby Point, D. C., and had charge of survey parties and dredging operations on several rivers and harbors.

In 1899 he was made inspector of reconstruction of Pier No. 4 Aqueduct Bridge, D. C., and later in the same year was placed in local charge, as assistant engineer, of the construction of the defenses of Washington, D. C., after which work he was made principal assistant engineer on all river and harbor, and fortification work. This position included much important work in connection with fortification, river and harbor improvements and municipal development in the District of Columbia.

At the time of his death Mr. Warman was chief assistant engineer with charge of the River and Harbor Work of the Fortification Division.

In college he was a member of the Tau Beta Pi fraternity and upon entering professional life became a member of the American Society of Civil Engineers and the American Institute of Electrical Engineers.

Mr. Warman died April 27, 1908.

SAMUEL WEBBER

Colonel Samuel Webber died at his home in Charlestown, N. H., on February 23, 1908 He was born December 9, 1823.

At 18 years of age he entered the employ of the Merrimac Manufacturing Company at Lowell, Mass., there developing the etching process for engraving rolls, and familiarizing himself with the operations of cotton manufacturing. He also assisted Dr. S. L. Dana in his experiments with boilers and fuel combustion. In 1847 Mr. Webber went to Lawrence, Mass., as draftsman and assistant engineer in building the former Bay State Mills. He was superintendent of these works in 1849–1850. In 1850 he was sent to Europe to study worsted and linen manufacture and to note the improvements in cotton machinery. While in London in 1851 Mr. Webber acted as one of the jurors on manufacturing machines and tools at the Crystal Palace, Hyde Park. Returning to this country in 1852 he set up a series of worsted machines in Lawrence and conducted extensive experiments with

them. In 1853 he went to New York and arranged the exhibition in Reservoir Square, acting as commissioner of juries. In 1854 he went to Springfield, Mass., and finished building a cotton mill at Indian Orchard, installing the machinery and operating it for four years as the Ward Manufacturing Company. In 1858 he went to Manchester, N. H., as manager of the Manchester Print Works, remaining there until 1864, when he resigned to take charge of the Portsmouth Steam Mill, manufacturing spool cotton. While there Mr. Webber in company with J. S. Davis of Holyoke, Mass., and Phineas Adams of Manchester, N. H., called the first meeting for the formation of the New England Cotton Manufacturers' Association. Early in 1861 he was commissioned colonel and aide-de-camp on the staff of Governor Berry of New Hampshire and was commissioned to equip and command the First New Hampshire Light Battery in May 1861. Colonel Webber went to Washington with this battery and the Fourth New Hampshire Infantry in October 1861, and turned over the troops to the United States government.

In 1865 Colonel Webber did a large amount of engineering work during the autumn and winter for the owners of the water power at Bellows Falls, Vt., measuring the flow of water there. In 1870 he collated the returns of the industrial division of the census at Washington. In 1871 he took up the question of the measurement of power, measuring by dynamometer the power used by cotton and other machinery, indicating steam engines, testing turbines, measuring waterflow, examining water privileges and acting as expert in power cases in court. Colonel Webber was one of the judges of the cotton and cotton machinery group at the Centennial Exhibition at Philadelphia and prepared many of the reports and data and was one of the judges at the Atlanta Exhibition in 1880. He was the author of many technical articles and he also wrote many interesting sketches of outdoor life which have been published in Forest and Stream, and other magazines.

ARVY ELROY WELLBAUM

Arvy Elroy Wellbaum was born February 12, 1881, at Brookville, Ohio, where he attended the high school. He studied at the Ohio Northern University, Ada, Ohio, for one year, and received the degree of M.E. in 1902 from Ohio State University. During the summer vacations he was in the employ of the C. & G. Cooper Co., Mt. Vernon, Ohio, and Platt Iron Works Co., Dayton, Ohio.

In 1902 he became draftsman for the Morgan Engineering Co.,

Alliance, Ohio. He becam econnected with the Foos Manufacturing Co., Springfield, Ohio, in 1903, as designing draftsman, and in 1905 he accepted a similar position with the Foos Gas Engine Co., Springfield. For three years he was instructor of mechanical drawing and machine design in the Young Men's Christian Association of Springfield. Up to the time of his death, August 31, 1908, he was associated with the Hydraulic Press Co., Mt. Gilead, Ohio, having had charge of the engineering department.

GEORGE W. WEST

George Washington West died at his home in Middletown, N. Y., December 24, 1908. He was born April 3, 1847, at Troy, N. Y., and received his early education in the public schools of that city. In 1865 he entered the service of the New York Central & Hudson River Railroad at Schenectady, as machinist, and was later made foreman and master mechanic, leaving this position to accept a similar one with the West Shore.

In 1886, he entered the employ of the New York, Lake Erie and Western, now the Erie, as master mechanic of the Mahoning division; was later transferred to the main shops at Meadville, Pa., and in 1888 to the Eastern division. From 1891 until the time of his death he held the position of superintendent of motive power of the New York, Ontario and Western.

Mr. West was past-president of the American Railway Master Mechanics Association, a member and past-president of the New York Railroad Club, and past-president of the Central Railway Club. He was a member of the Masonic order and the order of Elks. The George W. West Association of Engineers at Carbondale was named for him. He was also a director of the First National Bank of Middletown, president of the Ontario and Western Savings and Loan Association, a member of the Middletown Club and a member of the Board of Water Commissioners.

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